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INDIAN JOURNAL OF POWER & RIVER VALLEY DEVELOPMENT

HYDRAULIC MACHINES SYMPOSIUM NUMBER

VOL. X NO. 9 SEPTEMBER, 1960 RUPEES TEN

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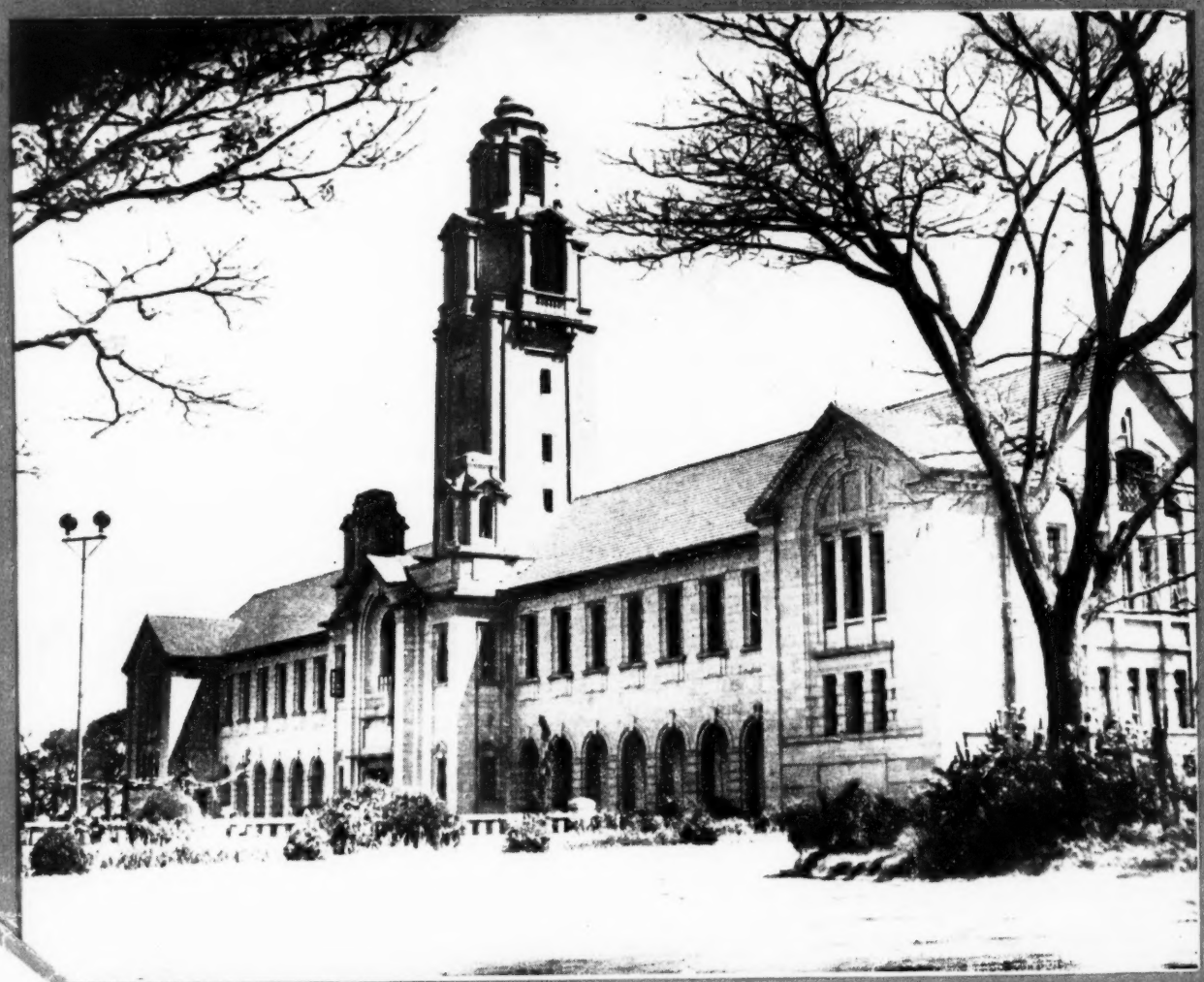

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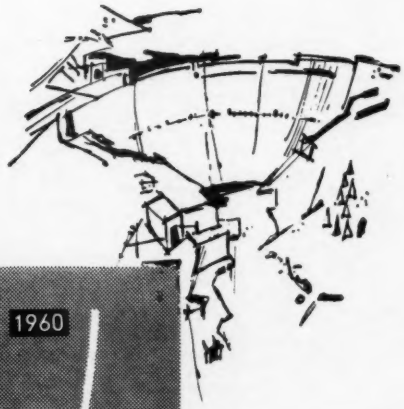

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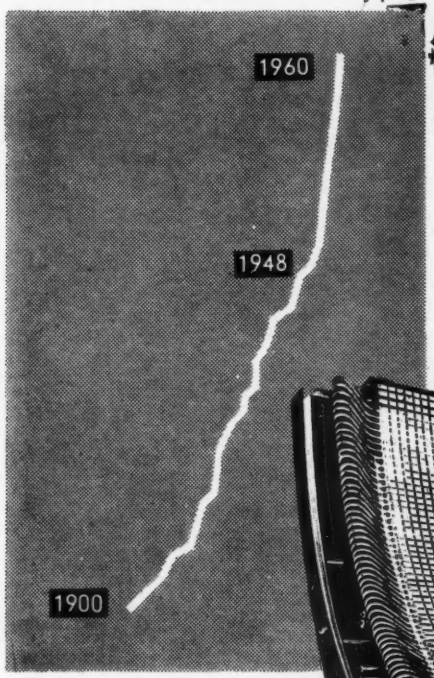

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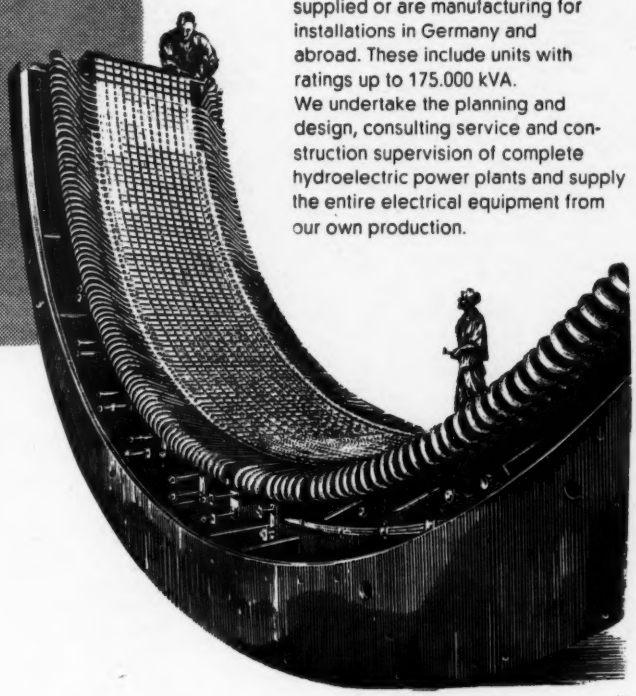

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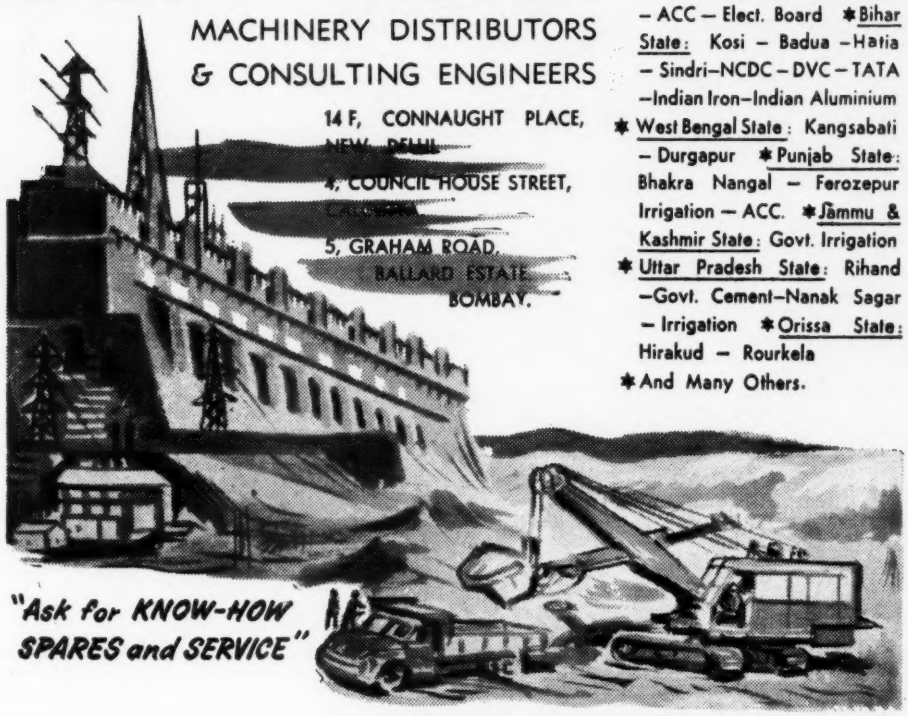

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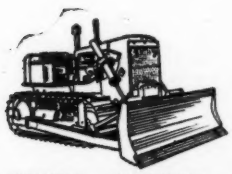

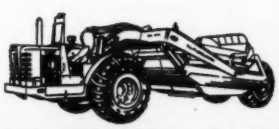

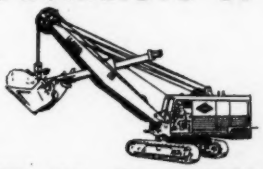

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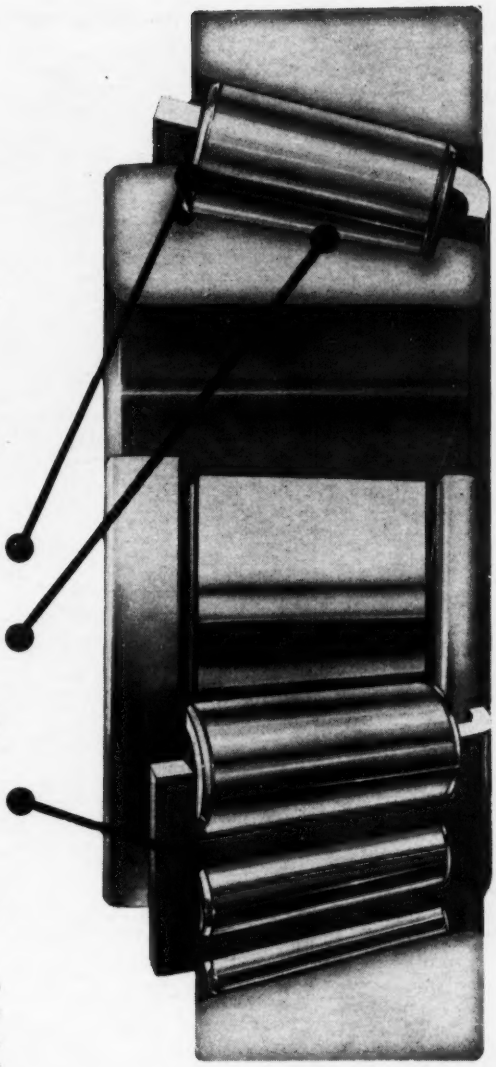

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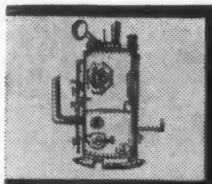
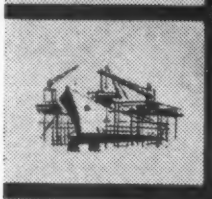
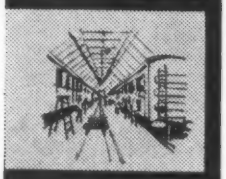
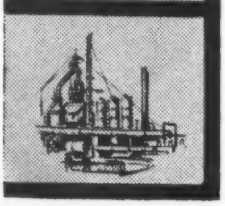

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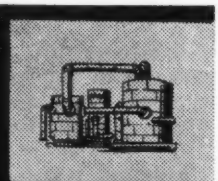
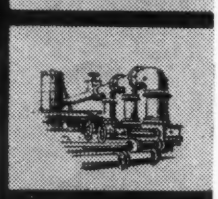
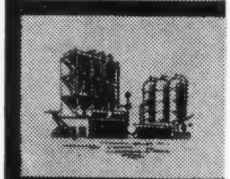

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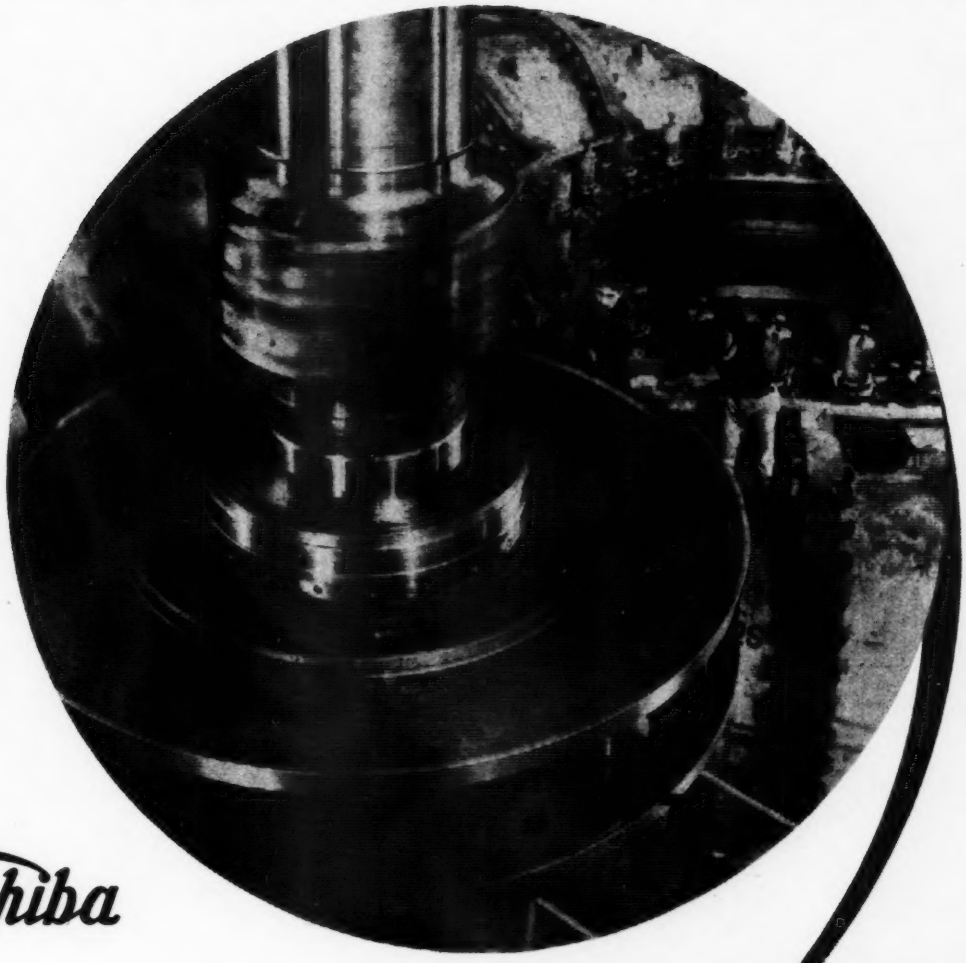

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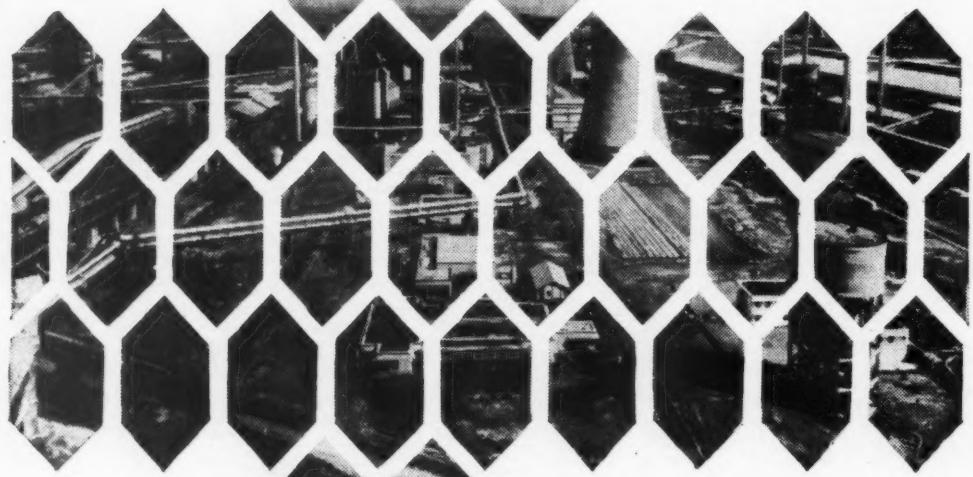

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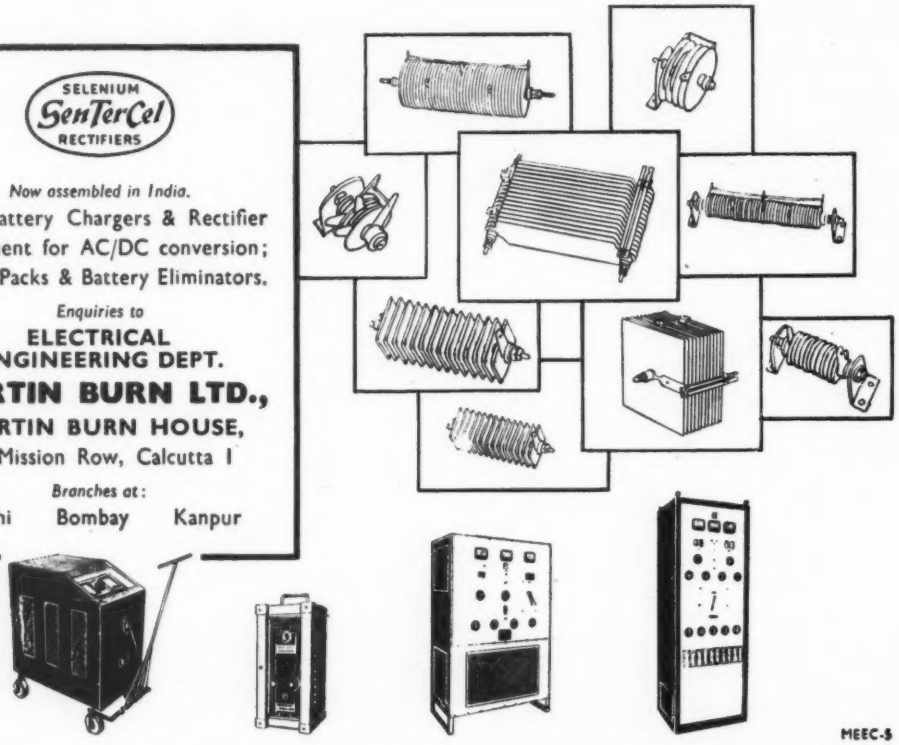

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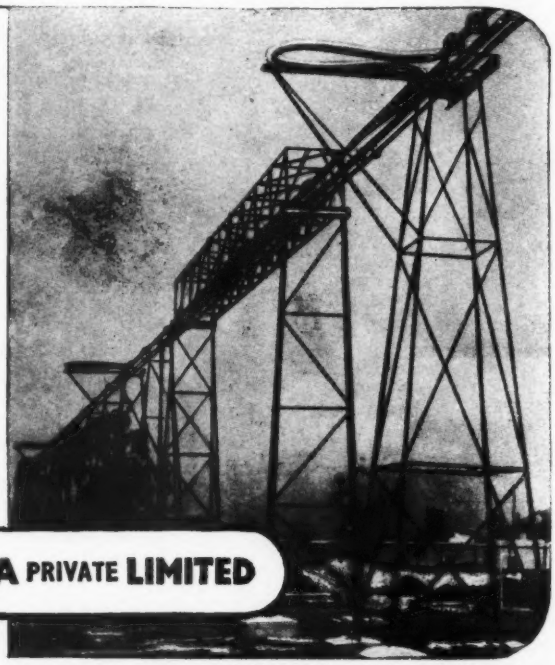

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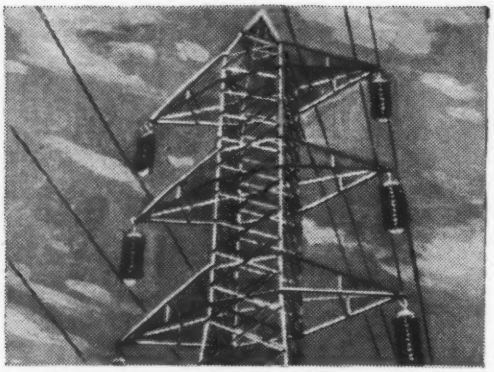

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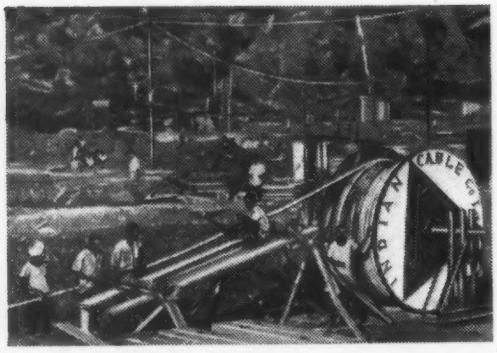

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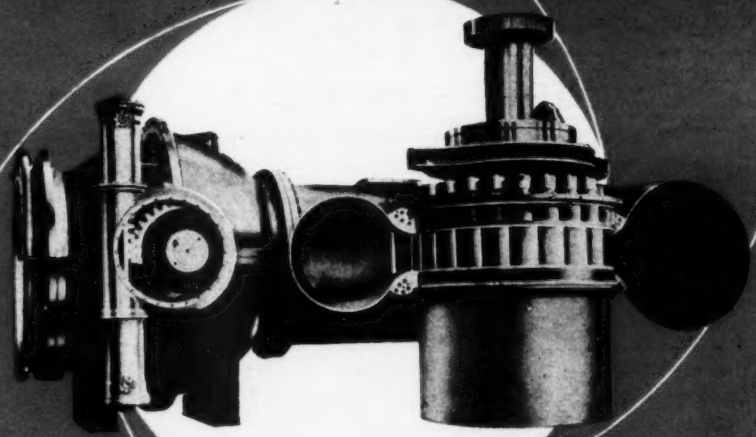

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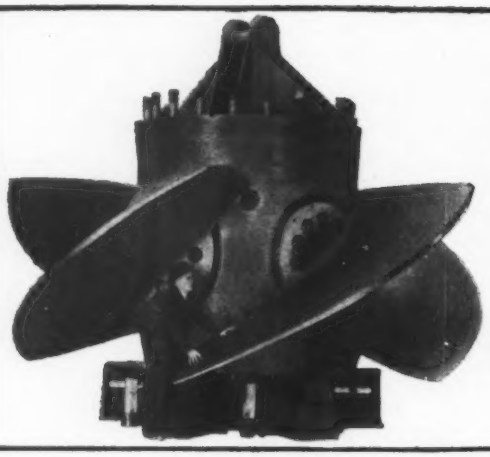

The outer head cover during manufacture. In the background you can see two Francis runners intended for The Stornorrfors Power Station, Sweden, each with a guaranteed max. output of 200,000 HP — *the most powerful water turbines in Europe.*

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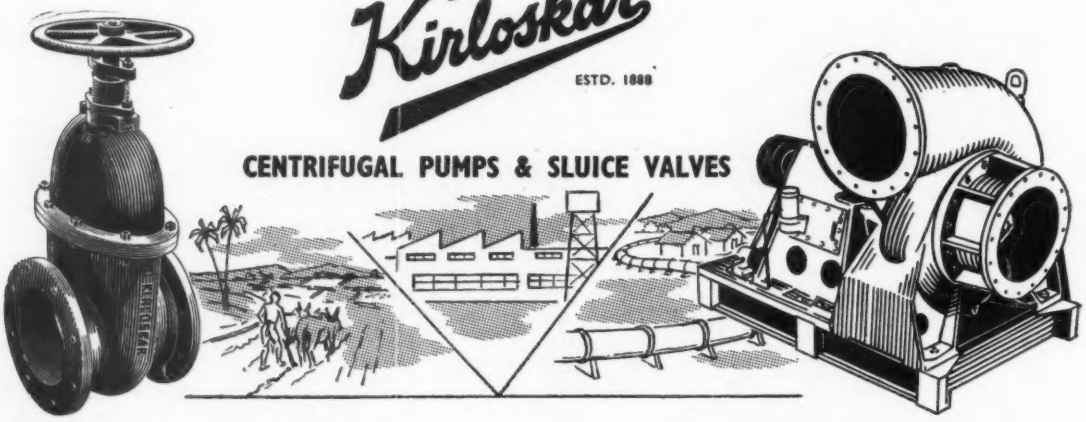

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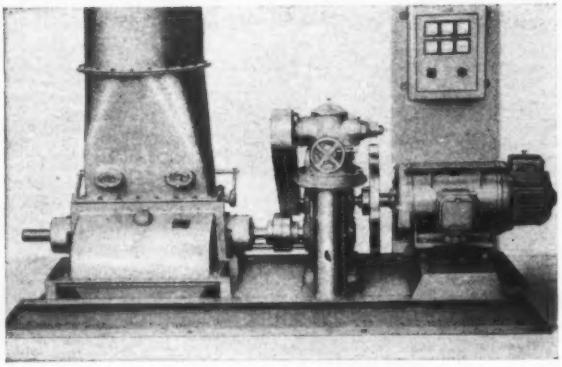

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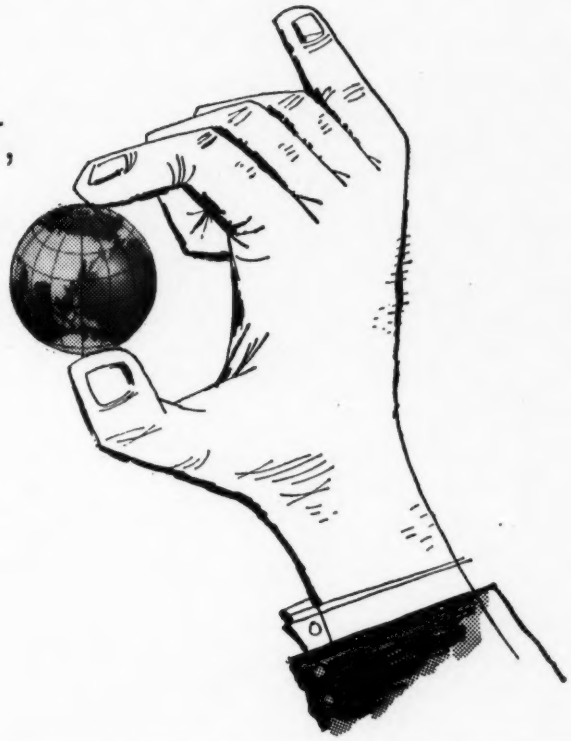

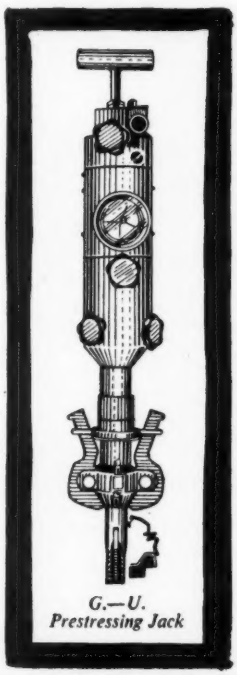

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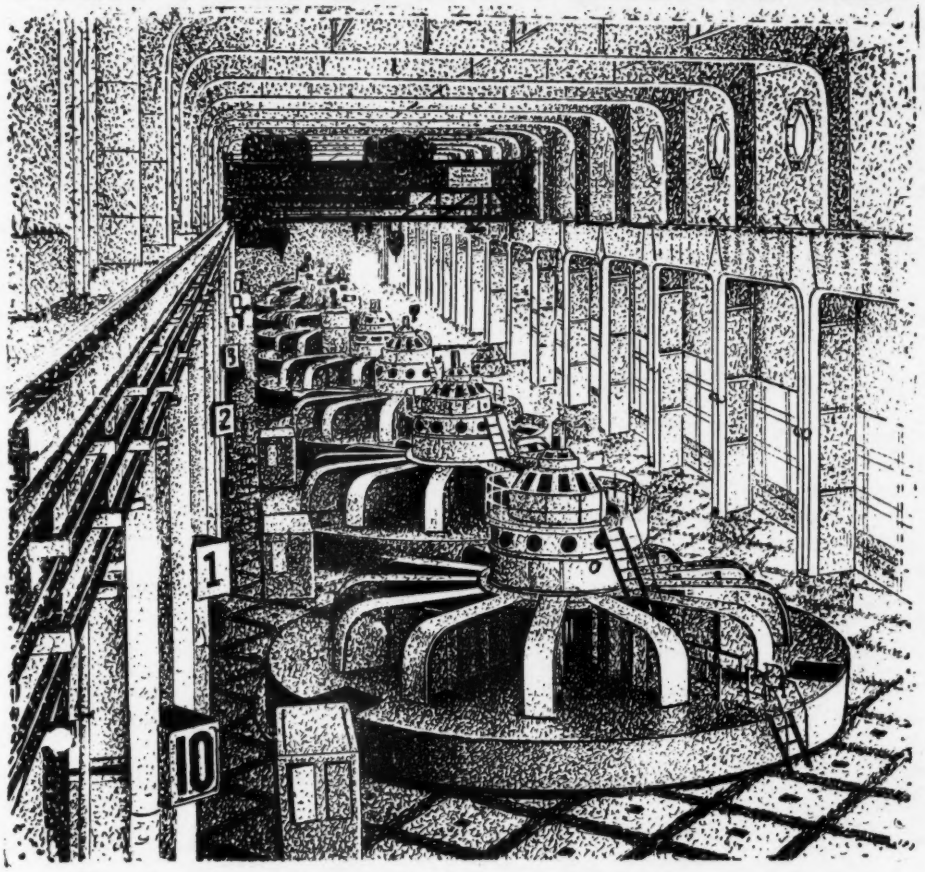

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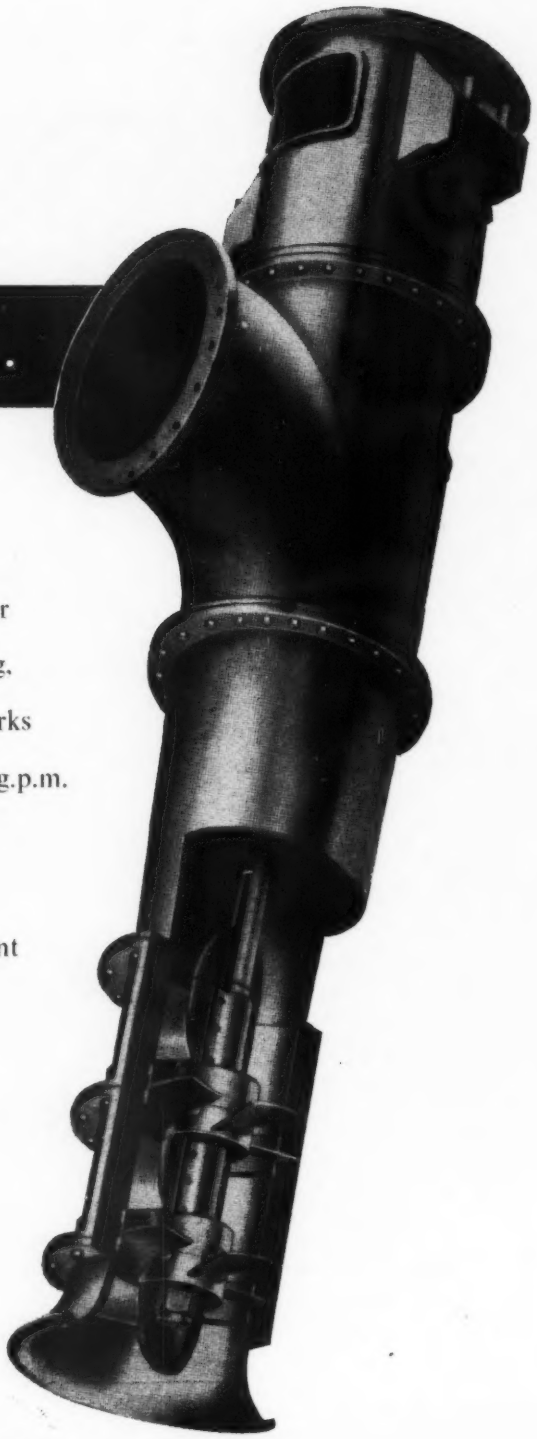

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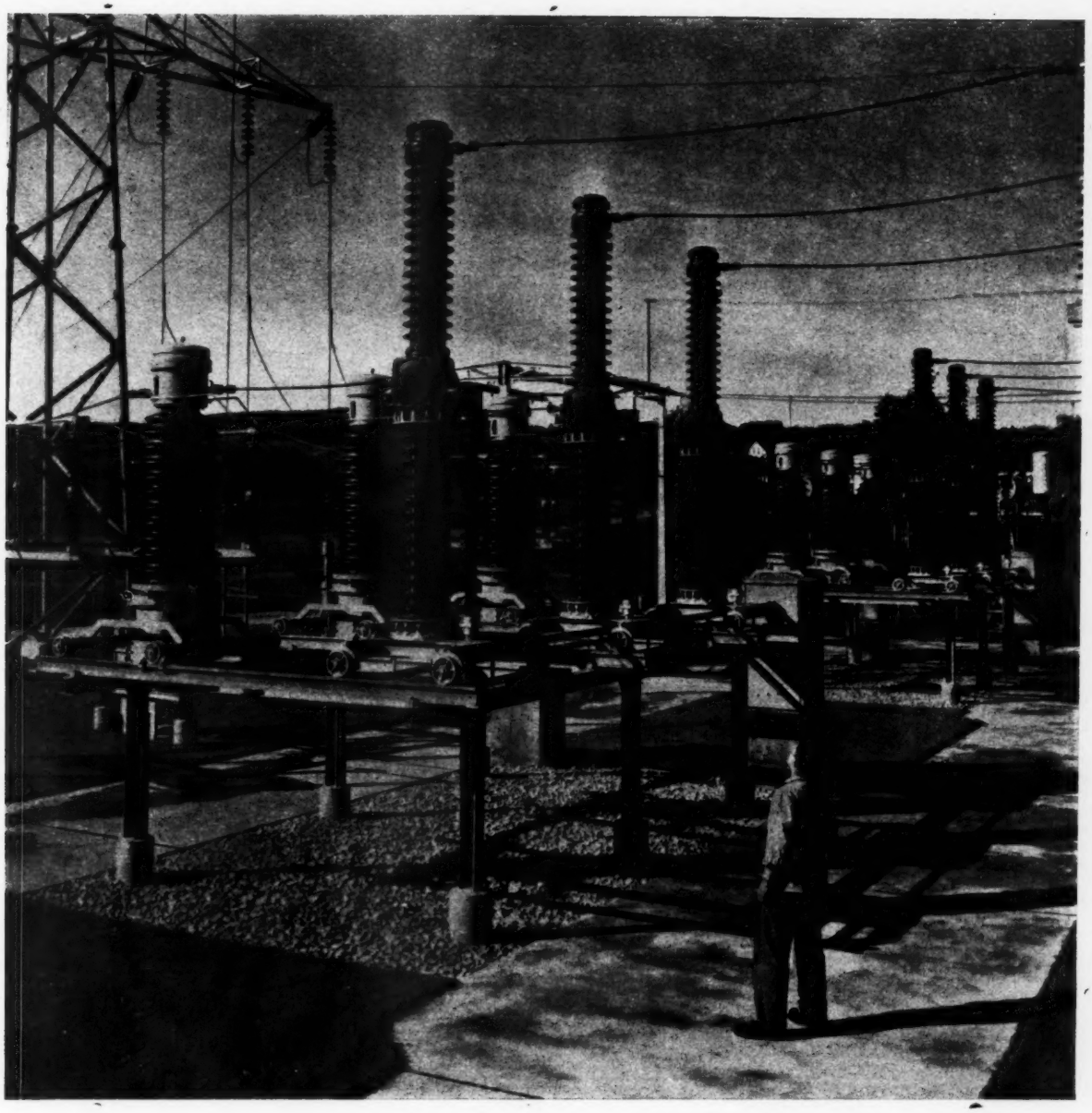

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VOL. X — **SEPTEMBER 1960** — **NO. 9**

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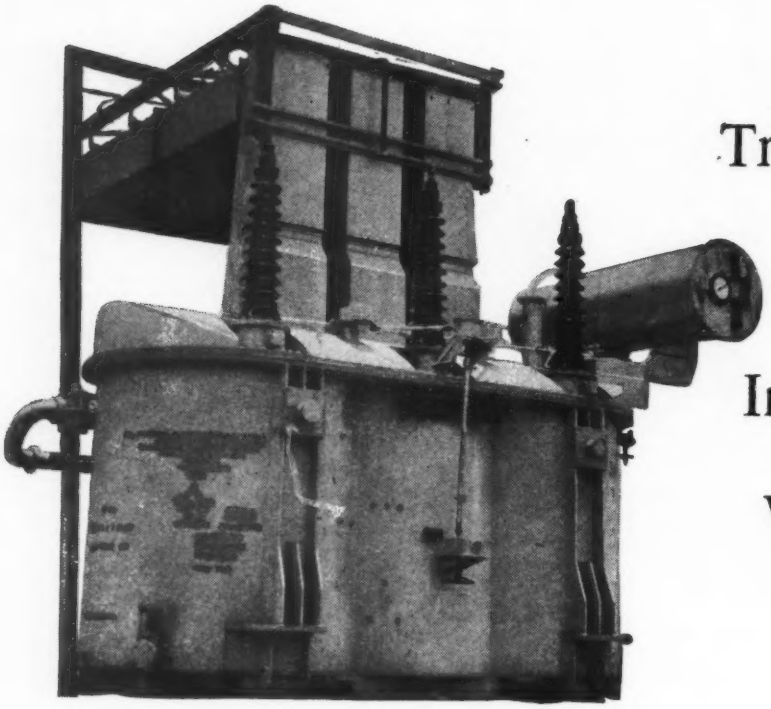

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Introduction

The Indian Institute of Science is one of the premier scientific institutions in the country which has contributed to research in various fields for over half-a-century now. It has also contributed substantially in training the scientific manpower of the country that are now holding leading positions in the various fields of government and industry. It was therefore but proper that its Golden Jubilee was celebrated in a befitting manner.

The present volume is the the result of the Seminar on Hydraulic Machines held at the Institute as part of the celebrations. It will be noticed that 48 papers were presented at the seminar and quite a number of them were by eminent engineers from abroad. We have selected 21 out of these for inclusion in this issue. These papers may be divided under the following heads.

Hydraulic Turbines
Pumps
Hydro-dynamics of Turbo Machines
Cavitation
Testing
Operation

We are confident that these papers will be found extremely interesting and instructive to the engineering profession.

We wish to express our thanks to the Civil and Hydraulic Engineering Section of the Indian Institute of Science for cooperating with us in bringing out this volume and, in doing so, we wish long life to the Institute and its continued service to the cause of science and engineering in the best of its traditions.

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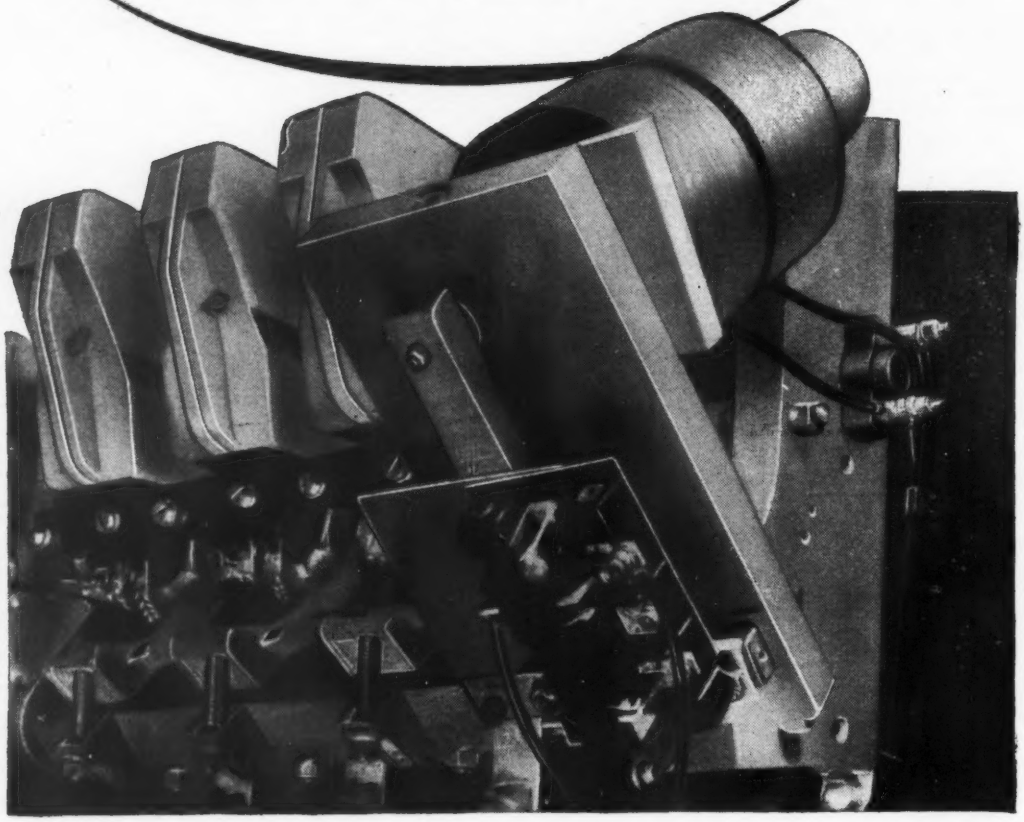

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A Warm Welcome

Being the Presidential Address by the Director of the Institute on the occasion of the inauguration of the Symposium.

By Dr. S. BHAGAVANTAM

The Indian Institute of Science has just completed 50 years of its existence. This important event, as many of you are aware, was celebrated in a manner befitting the occasion in February 1959. That was mainly the festive part of the show. Since this is an academic Institution whose main function is the development of research and learning; for that purpose, the whole of 1959 is being regarded as the Golden Jubilee Year. During this year, more than a dozen symposia are to be held by the various Departments of the Institute. More than half-a-dozen have already been held and the remaining are to be held during the next few months. The Golden Jubilee Volume consisting of research work currently done at the Institute is also to be brought out shortly. These activities constitute the academic portion of our celebrations. As part of these celebrations, this symposium is being held today.

Sir, we are extremely grateful to you for having so kindly consented to participate in our function inspite of your other multifarious engagements. I also wish to express my thanks to all the delegates who have made it convenient to attend this function inspite of their other engagements. I am fully aware of the extent of sacrifice each of you had to make in coming over here to participate in this function of ours. This symposium will have more than justified itself, if at the end of it, some of you at least feel that your sacrifice in coming over here has been worth your while. We are very grateful to you for the many papers that you have presented for being read here.

I have no doubt, the workers in the Civil and Hydraulic Engineering laboratories will be greatly benefitted by this meeting. I learn that the various bodies interested in the research, development and manufacture of hydraulic machines, the Indian Standards Institution; the Development Council of the Government of India on Pumps and Internal Combustion Engines; the All-India Pump Manufacturers Association; and the Users of hydraulic machines like the Chief Engineers of States, Professors of Engineering Colleges, have all contributed to the success of the symposium. My thanks are due to all these organisations and individuals. My thanks are also due to the Editor of "The Indian Journal of Power and River Valley Development" for having kindly agreed to bring out a Special Number of the Golden Jubilee Symposium on "Hydraulic Machines".

I understand that at present, most of the units manufacturing various hydraulic machines in this country are entirely depending for their designs on foreign consultants. This dependance cannot obviously go on for ever. It is necessary that a net-work of development and design units are established all over the country. Each one of these units should consist of personnel who are capable of designing the equipment necessary and act as consultants for initiating various development projects in the country. If we do not have such persons in sufficient numbers, we shall have to prepare them. This may be the hard way to learn but it is the only way by which teams of competent consultants can be built up within the country and industrial progress assured.

At the Indian Institute of Science, work on the subject matter of this symposium is being done in a modest way. Members of our staff have developed a current meter for measuring discharges in rivers, a water supply meter for measuring consumption of water in domestic buildings, a high discharge low head centrifugal pump for meeting the requirements of small agriculturists and so on. Development work is going on in the design of turbines and axial flow pumps. Given suitable encouragement, it should be possible to develop this laboratory into a first rate organisation for hydraulic machines.

I learn that some of the machines in our Laboratory have been specially designed for undertaking model testing of turbines and pumps. At present, hydraulic machine turbines are being purchased in this country depending entirely on the manufacturers' guarantees and the results of model tests furnished by the manufacturers' themselves. Even from a purely commercial point of view, it is not desirable that both the manufacturing and the certification of its quality should be in one and the same hands. The agency to test the quality should be different from the supplier. I would, therefore, like to point out that whenever big hydraulic machines are purchased, they be pumps or turbines, one of the conditions to be stipulated should be that manufacturers should supply models of the same. These should be tested according to internationally accepted standards in an Institute like this for their performance. The advantages of such a procedure are obvious and need no elaboration.

The tempo of industrial activities in the country has increased enormously during the past ten years and is bound to increase many times more in the coming years. That this Institute and its staff are not only competent to play their part but look forward with great earnestness to do so in a large measure needs no special mention by me.

Before I conclude, may I, Sir, extend to you and all the delegates again a very warm welcome to the Indian Institute of Science.

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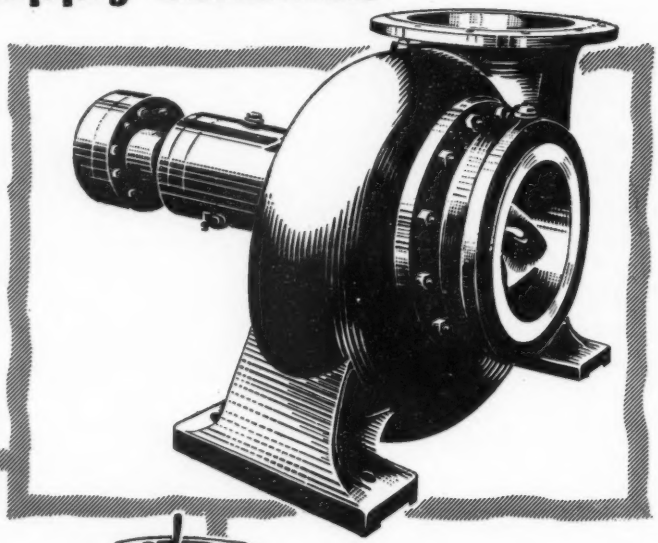

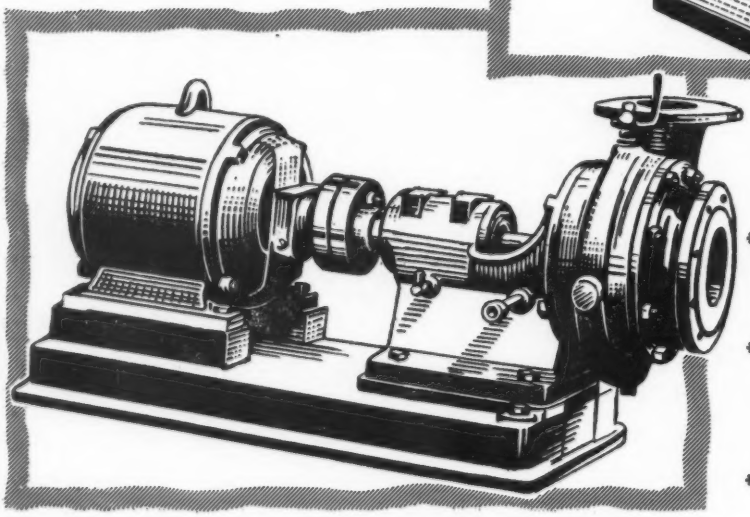

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Only Research can meet the Challenge

Being the Inaugural Address on the occasion

By MANUBHAI SHAH

From a modest start, this Institute has, within the last 50 years built itself up into one of the leading Institutions of its kind in the country. Its alumni drawn from all parts of the country have played notable part in the country's scientific and industrial advancement in all aspects—whether it be in helping to formulate the policies of the Government or in the laboratories in the various industrial concerns in the country.

At this stage of our industrial development, when we have to achieve maximum production with minimum of equipment, personnel and foreign exchange, it is essential to maintain close co-ordination between research and development. Your efforts, therefore, to hold symposium on hydraulic machines is welcome. Your suggestion to make greater use of leading Institutions such as yours in the designing and testing of hydraulic machines—turbines, pumps and other machines—is fully in line with the Government policies in this regard and I can assure you that any concrete suggestions that you would like to make in this connection will be most gladly considered by us.

Hydraulic machines cover a wide range of equipments and is used for diverse purposes such as agriculture, irrigation, sanitation generation, of power, etc. You will be glad to know that successful indigenous development has been achieved in the field of agricultural machinery in that we are now manufacturing in the country a variety of pumps and machines, which are capable of meeting the normal needs of the agriculturist.

It is in this field of hydraulic machinery, I am glad to say that we have made substantial progress. I would like to review this briefly.

In India the organised production of the centrifugal pumps was established in the year 1925 by one or two firms. The pioneering firms had to struggle hard to exist in the face of heavy competitions from pumps of foreign origin which were being freely allowed to be imported into the country. It had also to overcome the prejudice against the local products. The industry however was able to make headway during the Second World War. The Grow More Food Campaign gave further fillip to the industry as the increased demand for pumps could not be met from imports and the foreign firms were yet to switch over to peace time production. These favourable conditions were further strengthened by the new encouragement given by the Government towards industrialistion since independence.

In 1950, there were only 8 organised units manufacturing pumps with a capacity of 35,000 nos. per annum and a production valued at Rs. 1 crore; by 1956 the number rose to 27 units with an installed capacity of 68,000 nos. and by the end of 1959 there would be 55 firms with a total installed capacity of 1,30,000 nos. per annum valued at Rs. 4 crores of production per year. Of these five firms are manufacturing Deepwell Turbine Pumps which are required for developing the underground resources in certain areas of the country.

The production of power driven pumps has been as under:—

Year	*Nos.*	*Value.*
1950	30,292	Rs. 0.88 crores
1956	46,881	Rs. 1.36 crores
1959 (Estimated)	85,000	Rs. 2·46 crores

I should add here that many of the pumps that are being manufactured in the country have been undertaken in collaboration with wellknown firms abroad; some of them are: Fairbanks Morse of USA, Johnstons of USA, Worthington of U.K. and Sulzers of Switzerland etc.

You will be glad to know that some of the types of pumps that are already being manufactured in India are being exported to Middle Eastern and Far Eastern countries earning us foreign exchange to the tune of Rs. 5 lakhs. Therefore the quality of the pumps manufactured in our country speaks for itself as they are meeting competition from other exporting countries.

Although the industry is well established in India, pumps for special chemical applications and of large sizes are still being imported into the country. With the expansion and diversification of the industry that is being implemented, I hope we would achieve not only self sufficiency in the next two or three years but also export a large number of pumps and hydraulic machines to meet practically the requirements of the export market in the neighbouring countries.

I am glad to inform you that for the manufacture of the pumps most of the components are made from indigenous material.

Manufacture of hydraulic machines used for power generation is planned in the Heavy Electrical Project at Bhopal with a capacity of 0.35 million kW's

per year on double shift and the biggest turbine will be 50 mega watts. Further capacity of 0.3 million kW is under consideration for turbines between 10,000 to 100,000 kW in another Project. Hydraulic turbines upto 100 kW capacity are being made by one of the indigenous firms. Gates of all categories such as sluice, flood, penstock, are also under regular production.

While the indigenous manufacture is proceeding satisfactorily, very little attention is being paid by most of the manufacturing concerns to research and development, which alone can in the ultimate analysis make the indigenous industry meet the challenge of their counterparts in other countries. The lead given by the Institute of Science in arranging this symposium on hydraulic machines is, therefore, in the right direction and I hope it will be a forerunner of many such gatherings to be held periodically in different parts of the country so as to stimulate wide interest in the problems that confront the manufacture, operation and maintenance of hydraulic equipment. The industry in our country should bestir itself to more and more built-in-research for all these important lines in addition to the development and research taking place in National Institute such as yours and our National Laboratories.

I would like on this occasion to give you an idea of our plans for the production of steel in the third Five Year Plan period. It may not be directly connected with the subject of today's symposium but you will agree with me that the development of all engineering industries including hydraulic machinery is closely related and dependant on the availability of steel in adequate quantities. The target for the third Plan under consideration is 10 million ingot tons of crude steel, or about 7.3 million tons of finished steel. Recently, criticisms have been levelled against this target in certain quarters on the ground that the consumption is not likely to come to these expectations and that unnecessarily a high target might divert to steel a large proportion of the total financial resources thereby curtailing development in many other fields. I feel that such apprehensions are groundless. Steel forms the basic raw material for all engineering industries. Steel and Power, as you know, are the foundation of industrialisation. During the Second Plan period, the development of all engineering and secondary industries have suffered considerably due to the nonavailability of steel, the import of which had necessarily to be restricted due to shortage of foreign exchange. Quota allocation of steel made in favour of actual manufacturers and fabricators more than 12 months ago have yet to materialise. Even the full utilisation of installed capacity—not to speak of expansion, modernisation or diversification of production—was not possible in many industries for want of iron and steel.

And yet we know that as against the annual general index of industrial production which is estimated at 148 points for this year, i.e., the year 1959 (with base year 1951 as 100), happily the estimated index for group of the machine building industries, which is the most important of the engineering industries, stands at 433 for the current year; for electrical machinery industries, it is well over 302 and for general engineering industries, the index has pushed over 273. You will be pleased to know that for machine tools industry, the index is running at 888. This is a very encouraging and promising trend of industrial production.

This only demonstrates the capacity and growth of steel consuming industries. If during the Second Plan period, iron and steel were more freely available, this production in the main steel consuming industries like machine building, machine tools and engineering industries and structural and secondary industries would have been much greater than even the very happy performance of these industries that we are witnessing today. The effect of the non-availability or the great difficulties in procurement of steel that our steel consuming industries and steel processing engineering industries have experienced in the last three years and which they are experiencing even now, are fully known to all concerned.

The production during Second Plan period was also handicapped due to the non-availability of foreign exchange to import capital goods, plant and equipment, steel, non-ferrous metals and basic raw materials in adequate quantities. A very appreciable portion of the total foreign exchange had to be diverted for the import of steel. These and such other factors have to be taken into account by the people, the Government and the Planning Commission in fixing the target for steel for the Third Plan. Even if there is a risk in planning for a somewhat higher target, I feel it is better to err on the side of a higher practicable target like 10 million tons of crude steel because any shortfall in production of steel in relation to the demand would have very wide repercussions on the pace of development of the vital machine building industries, heavy engineering industries, basic industries and vital light engineering consumer goods industries. Prime pumping of steel in the National Economy without the elaborate and cumbersome distribution controls like the present Iron & Steel Controls, would revitalise and strengthen the national economy and industrial development in a very healthy and dynamic manner.

Therefore, I do hope that Symposia and Seminars like yours would explain and clarify to the public the vital and urgent need to think more realistically and far-sightedly about the production targets of steel in the Third Five Year Plan. If we make a mistake again of being unduly conservative on steel production, the nation and its industrial development will suffer greatly. I pray and plead that we should always be a little surplus, if at all, in steel if we have to advance rapidly in the industrial field. I am fully confident that in the Third Plan, with the great emphasis that we are going to give to machine building and engineering and basic industries, we could easily consume 10 million tons of crude steel or 7.5 million tons of finished steel and even a little more per year by 1965-66. We should resolutely plan for this target.

Review of the Papers

Being a brief review of papers presented at the Symposium.

By N. S. GOVINDA RAO

Altogether 48 papers have been received for the symposium. Contributions are from India, USA, United Kingdom, France, and Germany. The contributors are professors of Universities, engineers in manufacturing industries and those in-charge-of execution and development of hydro-electric or pumping projects.

Hydraulic Turbines

Sixteen papers have been received on the subject of Hydraulic Turbines.

Sri V. P. Appadurai has reviewed all aspects concerning the design and development of water wheels, their selection, construction, operation, installation and maintenance. He has given an exhaustive table of turbines manufactured and in use together with the details of their various characteristics. These will be very useful for selection of turbines in future projects.

Sri A. Thiruvengadam has also dealt with problems of designs and development of turbines. He has reviewed the various developments that have taken place in design and construction of turbines. He considers that future developments in hydraulic turbine designs practically depends on further research on cavitation, corrosion and erosion.

Survey of the materials used on the construction of turbines has been made by Sri V. P. Appadurai. Both Sri Netsch and Sri S. Padmanabhan have dealt with the design aspects of draft tubes. Sri Netsch deals with the investigation concerning the unusually long draft tubes for each of the four 1,35,000 HP Francis turbines on an under-ground power station in Australia.

Sri Padmanabhan deals with the function of draft tubes, their different tpes, and the methods of their design. He stresses the importance of model studies before finalising the design of a draft tube.

Messrs V. P. Appadurai, and P. Govindakrishnayya deal with the maintenance aspects of hydraulic turbines.

Sri Govindakrishnayya has stressed the importance of using proper lubricants in the maintenance of hydraulic turbines. These lubricants can only be selected after a clear understanding of the functions of the various parts of hydraulic turbines which rotate and the methods provided for their lubrication.

Regarding new types of turbines, Sri Deriaz and Sri Danel the original inventors have contributed papers. Sri Deriaz has described in detail the design of the Deriaz type of turbines, which is the mixed flow Francis type. According to him, Deriaz type turbines is best suited for heads higher than 150 ft. He has shown by comparison that for such heads, his turbine is superior to Kaplan in every way.

Sri Danel has given a very detailed description of the bulb type turbine, which has great possibilities of use in the development of power for heads less than 35 ft.

On manufacture of model wheels, Sri R. K. K. R. Govindarajan and Sri S. Nagaratnam have described the design and manufacture of a model Pelton wheel which is comparably good in efficiency and performance with a similar prototype.

There are a few papers on newer types of power development schemes. Sri N. Rajaratnam describes in detail pump storage schemes and their utility for peak load service. The design aspects of turbine-cum-pumps suitable for this purpose have been discussed.

Sri Danel has described the use of bulb turbines in development of power in tidal estuaries and in irrigation channels and anicuts across rivers etc. The possibilities of such development of tidal power has been exhaustively treated by Sri G. Suryanarayana Iyer and Sri A. S. Nagaraj in their paper. They have stressed the history of tidal power development and described a few types of tidal plants, such as the pit type units, the bulb type units, and the impact water wheel types. The design aspects of civil engineering structures, in such projects have also been briefly described.

Pumps

There are 16 papers. The design and operation of axial flow pumps have been described in the papers of Messrs N. Dwarakanath, N. S. Govinda Rao and von Ing. K. Rutschi.

Sri N. Dwarakanath describes the development of research in axial flow pumps. He considers that though design by the modified air-foil theory may be a small improvement on the existing methods, most rational method can be achieved when a three-dimensional theory of flow through turbo-machines has been analytically developed.

Sri N. S. Govinda Rao has described in detail a new theory for designing axial flow pumps. He has described the characteristics of impellers designed to produce vortices which have a function between a

free and a forced vortex. Sri Rutschi has described further research work necessary on axial pumps in this paper.

Regarding Centrifugal Pumps, Sri Kessler has stressed the historic development of the centrifugal pumps from 1754. He has also tried to classify the centrifugal pumps based essentially on specific speeds. He has also given description of the various prime movers, operating difficulties and maintenance of pumps.

The problem of large pumping installation has been dealt with by Sri Igor J. Karassik. He favours single suction centrifugal pumps with vertical motors. He quotes the example of the 65,000 HP pumps installed at Grand Coulee and 1,02,000HP pump installed at Hywasee in support of his contention. He has presented another very interesting paper on finding the various characteristics of a pump when its hydraulic details are not furnished by the manufacturer.

Sri M. W. Lalchandani deals with the input aspect of the centrifugal pump. He stresses the importance of correctly determining the pumping input at the design point and to so arrange that the maximum input anywhere throughout the range design does not very much differ from the input at the design point.

Sri Netsch's paper summarises the results of tests on centrifugal pumps of both the radial as well as the axial type with reduced runner diameters. The effect of reducing diameters has been thoroughly studied.

Sri Rama Murthy describes the various factors governing the mechanical design of centrifugal pumps. The nature of the loads and stresses which are to be considered in the design of the various components have been discussed.

Sri Rutschi has contributed a paper on modern centrifugal pumps and also on design of pumps with technical simplicity and economy. Another paper of his deals with the comparison of efficiencies when hydraulic machines operate singly and in groups.

Sri Keshava Murthy has described the design of low-head high discharge centrifugal pumps mainly for small irrigation requirements. The delivery head required for irrigating about ten acres of wet land is hardly a foot. Centrifugal pumps on the market have minimum delivery heads very much in excess of this requirement. Axial flow pumps require large capacity motors making them costly. This paper describes a pump designed solely to meet the requirements of small irrigation.

A paper by Sri N.S. Lakshmana Rao describes the studies made on air entraining vortices in pump sumps. He has also described the various methods of preventing formation of vortices in such pumps.

Testing of all pumps has been dealt with by Sri K.R. Pattabhiramaiah in his paper.

Hydro-Dynamics of Turbo-Machines

Many interesting papers have been contributed on the Hydro-Dynamics of Turbo-Machines.

Papers by Sri N. V. Chandrasekharaswamy, Sri Netsch and G. A. J. Young describe the nature of flow in hydraulic machines. The nature of turbulence characteristics has been described by Sri Chandrasekharaswamy. The flow conditions in turbomachinery and their influence on pipe lines of pumping stations and power plants, have been described by Sri Netsch.

The nature of flow in rotating passages of centrifugal pump impeller has been described by Sri Young. Some authors have tried to derive parameters for simulating flow conditions in turbo-machinery. Sri Netsch has one paper on this subject, while Sri Rutschi has another on the same subject. Sri Rutschi has tried to correlate Reynolds Number with dimensionless co-efficients of flow machines.

The effect of fluid friction of flow in hydraulic machines has been described by Sri D. S. Srinivasa. He has discussed the modification necessary in the Euler's equation to take this into account. Further research is necessary on the velocity distribution in the impeller passage in centrifugal pumps. In axial flow pumps C_L should be calculated taking this effect on separation of flow into consideration.

Cavitation

The cavitation and steps necessary to ensure similarity in cavitation effects between model and prototypes have been presented in the papers by Sri B. P. Singhal, Professor A. J. Acosta and Professor A. Hollander, Sri K. Seetharamiah and Sri L.H. Kessler.

Sri B. P. Singhal has discussed the various stages in the development of cavitation bubbles and scale effects in model testing of hydraulic turbines.

Professor A. J. Acosta and Professor Hollander have reviewed the development of the various stages of cavitation in a machine. They have followed this by a discussion on the various dimensionless parameters used to describe cavitating performance in a turbomachine. Taking a free streamline model of flow in a simplified impeller, they have made theoretical estimates of the extent of cavitation as a function of blade geometry and angle of attack. Some recent work on the effect of thermodynamic properties of the fluid on cavitation performance has also been discussed.

Sri Seetharamiah's paper deals with the physical causes of cavitation, classification of the phenomena to several types and to the causes of inception of cavitation. He has indicated that in predicting cavitation effects on prototypes from model studies, it should be remembered that turbulence is not scaler and the time required for a bubble to travel before collapse, is the same both in the model as well as in the proto type. He has also suggested various methods of preventing cavitation. Sri Kessler has described in his paper the method of designing pumps to avoid cavitation conditions.

Testing of Hydraulic Machines

Six papers have been presented on the Testing of Hydraulic Machines: Sri V. P. Appadurai has described in his paper the general methods of testing water wheels.

Sri Netsch has described the research on stability of Pelton turbines. He discusses the relative advantage of having a single or a dual control system for operating the needle governing movement of the jet in a Pelton turbine. Generally for Pelton's of smaller type a single controller is enough. Sri Rutschi describes the stabilization of efficiencies of pumps and turbines. Sri Pai has described in detail the method of testing turbines.

Sri R. K. K. R. Govindarajan's paper deals with description of universal hydraulic test rig of 15 HP and 5 ton capacity. With the help of this, it is possible to conduct a variety of tests on differnt pumps and turbines.

Sri Norman R. Gibson has described in great detail his experiences in the use of Gibson's Method of water measurements for efficiency tests of hydraulic turbines.

Operation of Hydraulic Machines

There are four papers on the operation of hydraulic machines. Sri V. P. Appadurai's paper deals with in a general way, the problems governing hydraulic turbines.

The considerations that have to be taken into account, in determining governor's capacity have been described by Sri Ravi. Sri Ravi has reviewed the principles of designing hydraulic turbines governors. He describes the advantages of using electronically controlled governors in preference to hydraulic controlled servo-mechanisms.

Sri Farrest Drake has also described the methods of automatic control of machines and processes. Methods of elementary speeds have been given. He has mentioned the characteristics necessay of a prime mover and of the speed sensing elements for afficient governing. Various methods of improving governor performance have also been discussed.

Sri Netsch has a paper on water hammer computations of ground discharge pipes by Laplace transformation.

There is only one paper on standardization of turbomachinery by Sri Rutschi. He has described in great detail the method by which various sizes of pumping machines can be standardised to cover a very wide range of flow and head conditions. The method adopted in Germany has been described.

Sri C. N. Nagaraj's paper deals with the method of designing of foundations of turbomachines. The foundations should be so designed that no resonance is set-up between the vibrations of the machine and the foundation-cum-soil system. A method of determining the natural frequency of such soil systems has been described.

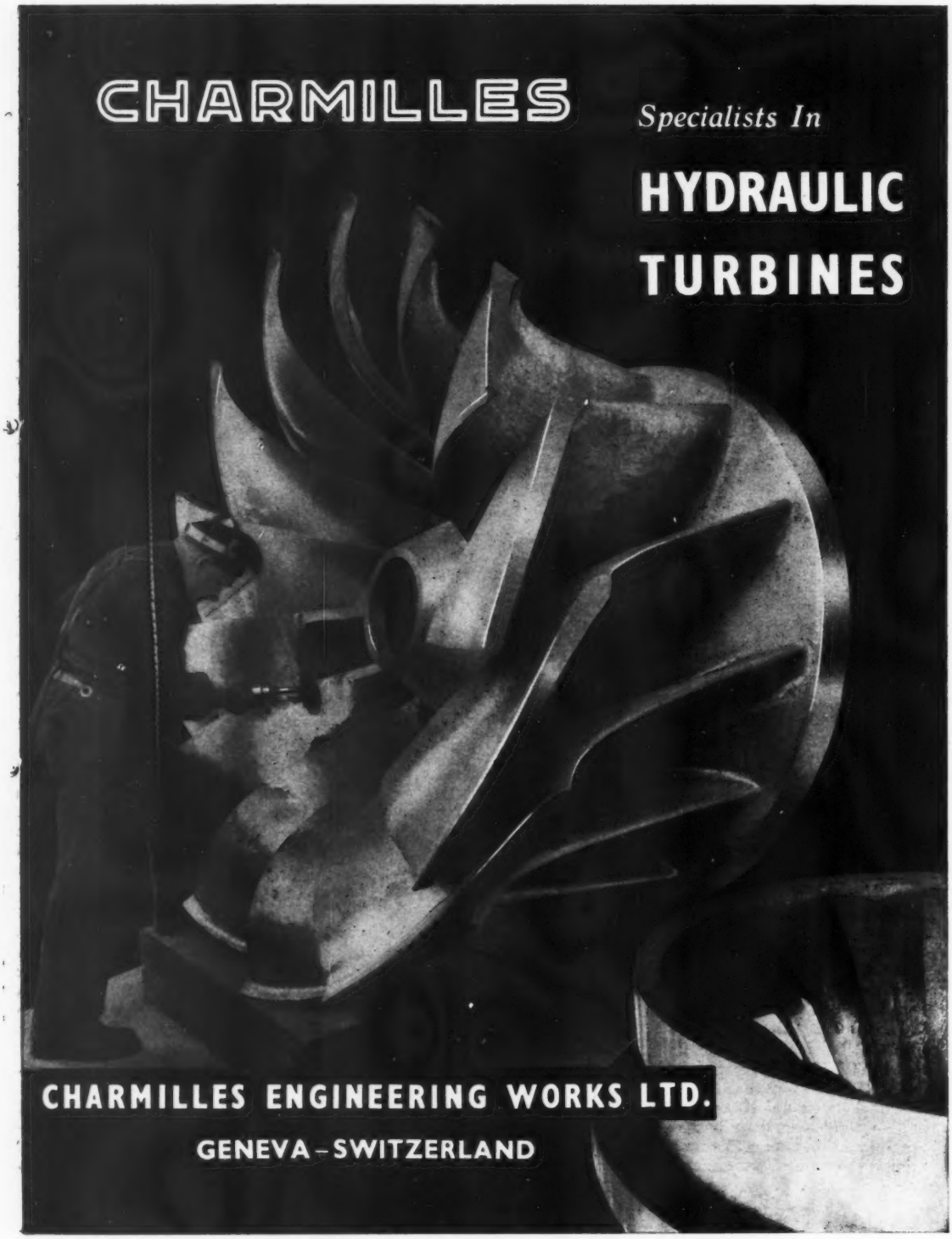
Sri D. Firth and Sri I. S. Pearsall have described great detail the effect of blade number on the torque/slip characteristics of a small fluid coupling. They have found the torque slip characteristics were found to depend upon the number of blades, particularly at high rate of slips. They have also given a semi theoretical method of analysis to make an approximate estimate of coupling performances.

From this brief survey it is evident that practically all aspects of turbines and pumps have been covered. There are hardly any papers on hydraulic brakes as in automobiles and aircraft, on fluid power transmission in automobiles, aircrafts, accumulators cranes, presses and a host of other devices in daily use. It is necessary that research and development work is oriented in these fields as well.

Regarding turbines, it is important that steps are taken to standardise acceptance tests of turbines by model testing in laboratories outside those of the manufacturer. If a design and development institute is started it whould not be difficult to supply design drawings for many of the turbines required for power generation. The States responsible for executing the hydroelectric projects or any central authority can get the turbines made in the country by getting the various components made in the several specialised workshops which are already in our country and assemble them in a particular place. It is necessary that the design of draft tubes are made in our hydraulic research stations which have a better knowledge of local conditions. Hardly any research work has been done in our country on the structural design of both the mechanical and civil engineering parts of a turbine—the cross tube, wicket gates, the foundations for the generators etc. A start has to be made.

Regarding pumps, practically all papers deal with clear water pumps of comparatively small sizes. More development work is called for in the design of pumps for heavy density liquids, liquid-solid mixtures, high temperature liquids, and for high heads. There is not a single paper on types of pumps other than centrifugal ones, like reciprocating pumps, vane pumps, jet pumps etc.

On the hydrodynamics of turbomachinery, considerable analytical and experimental work has still to be done. Analytical work on three-dimensional analysis of flow, determination of C_L taking friction into account, nature of turbulence, cavitation, movement of solids in solid-liquid mixtures etc. has still to be continued. Since most of hydraulic machine design problems have to depend on empirical constants, experimental work is absolutely necessary if we have to progress in this field. Experimental work should not only cover finding design constants, but also fundamental research on problems like those of cavitation, turbulence, pressure and velocity distribution in impeller passages losses in bends, valves etc.



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Water Wheels

By V. P. APPADURAI

The hydraulic turbine in its present state of development is almost a perfect prime mover for central station service. It is efficient, of simple construction, and easily controlled. Its ability to act as a peaking or standby unit is outstanding, as it can be started cold and assume full load in a matter of minutes, follow load variations with little attention, going from no load to full load in a few seconds and can drop load without damage. Due to its simplicity, it can be made fully automatic and operate for weeks without attention. When the hydro-electric plant is combined with steam plants, in the same system, the steam plants usually carry that block load and the water turbine the swings in the system.

RECENT DEVELOPMENTS

A general survey of the hydraulic turbines manufactured in the past ten to fifteen years indicates that no major invention has been made recently in this particular field of engineering. Development has been concentrated on improvement of efficiency greater output, higher specific speed and upper limit of head for each type of turbine. All the turbines produced still belong to one of the three main types of machines, namely, Pelton, Francis or Kaplan. It appears that the Propeller type turbine has been ousted completely by Kaplan unit and speaking generally of large turbines, it can be said that nearly every turbine, manufactured in Europe now-a-days for a head of less than, say, 100 feet, is a Kaplan turbine and only very few indeed are propeller or Francis turbines. This is due to the fact that Kaplan turbines, because of their smaller dimensions and higher speed of rotation are hardly more expensive to install than propeller turbines and are therefore preferred, even when the head and discharge variations are so small, or the number of units so large, that very little gain in power generation is obtained by the use of Kaplan turbines. The fact that the maximum efficiency of Kaplan turbines is higher is itself sufficient justification for preferring this type of turbines.

Head

The maximum head for which each class of turbine is used has increased gradually, the maximum advance in this respect having been made with Francis turbine for which the head has risen from 1060 to 1710 ft., in the same period the advance had been 25% for Kaplan turbines viz., 184 to 230 ft. The advance in Pelton turbines has been only 2 per cent, viz., 5,700 to 5,800 feet, solely because there have not been higher heads to equip.

Two of the high head stations with Francis wheels are Hemsil in Norway with a head of 1,680 ft. and Ferra power station with a head of 1,710 ft. (under manufacture).

The higher heads for which Kaplan have yet been constructed is the 230 ft. at the Best Rhue Plant in France.

The upper limit of head for a pelton wheel is determined solely by the ability to obtain adequate strength of buckets and wheels; the highest existing head at present used is 5,800 ft. at Reisseck in Austria. Over the years, experience has taught that for satisfactory performance and long life, specific speed and head are related within fairly ingrossed limits. The experience curve is given in Fig. 1.

In installations of small horse power ratings, and where abrasive materials are carried in the water, the impulse type of turbines may be selected for head as low as 400 to 500 feet.

Efficiency

The turbine efficiency has been improved step by step and has reached the present level which appears to be about 93% for Kaplan, 92% for Francis and between 90 and 91 per cent for Pelton turbines.

Output

Turbine outputs have been gradually stepped up to the existing limits of 200,000 H. P. for Francis wheels at the Chute-des-passes station in Canada, 131,000 H.P. for Kaplan wheels at Priest Rapids in the U. S. A. Single jet twin runner pelton turbines of 150,000 H.P. are to be installed for Cimeys Power Station in the Italian Alps and 4 jet single runner pelton wheels of about the same output have been installed in the Kemano Power Station in the British Columbia. This represents a considerable advance in output of impluse turbines. The continuance of this upward trend in capacity is a matter more of economics than of any technical or design problems. Although the cost per H. P. of the whole machine will normally be reduced as the capacity increases, there is a limit beyond which the reverse will result and which can only be determined in each particular case. The optimum capacity is affected by problems of transport to site and erection and some extent by the particular manufacturer's production facilities. It is probable that there will be many future installations where such large capacity machines as those mentioned above are economically or operationally suitable and it is unlikely that there will be much increase above these outputs.

Turbines are selected to operate at a load factor of not less than 50% of the rated capacity for multipurpose projects and not less than 60% for run-of-river developments.

CLASSIFICATION OF TURBINES

One of the first problems confronting the designer of the hydro electric plant is the selection of the type of the turbine for the given operating conditions. Modern hydraulic turbines can be broadly classified as impulse and reaction turbines.

Reaction turbines may further be classified as axial flow and inward flow turbines. The runner vanes on the axial flow turbine may be fixed or adjustable; the fixed vane turbine is preferred when head and flow are substantially constant and where base load operation is possible. The adjustable vane turbine is preferred when these factors vary over a wide range. The inward flow turbine being stronger is better fitted for higher heads. Also the discharge velocity is relatively high permitting higher setting above tail water level without running into cavitation difficulties.

Reaction Turbines Fig. 2

In the reaction type, the water enters from a casing or flume with a relatively low velocity, passes through guide vanes of gates located around the circumference and flows through the runner from which it discharges into a draft tube seated below the tail water level. All the water passages are completely filled with water, which acts upon the whole circumference of the runner. Only a portion of the power is derived from the dynamic action due to the velocity of the water, a large part of the power being obtained from the difference in pressure acting on the front and back of the runner buckets.

In a reaction turbine, runner receives the water under pressure in a radial inward direction and discharges it in a substantially axial direction owing to the rotation of the runner, a centrifugal force is imparted to the water which opposes the inward flow.

The pressure at the runner inlet must be capable of overcoming the centrifugal force. Conditions somewhat resemble those of a D. C. motor where the rotation causes a back E. M. F. In addition, the pressure must produce the required acceleration of the water within the runner to discharge it through the tapered water passages by the vanes.

The fact that the water pressure drops within the runner identifies the Francis turbine as a reaction turbine. The degree of reaction is defined as the ratio of the pressure difference between points A and B and the maximum head under which the turbine operates.

The draft tube allows maximum utilisation of the available head, both because of the suction created below the runner, by the vertical column of water and because the outlet of the draft tube is larger than the throat below the runner, thus utilising a part of the kinetic energy of the water leaving the runner blades.

In other words, the reaction turbine obtains rotative force by deflecting the flow of water in closed passages and utilising the consequent reaction.

Impulse Turbines

In the impulse turbine, the water is open to the atmosphere at all points beyond the nozzle and the transfer of energy from the water to the turbine runner is due to impingment of the jet of water on the buckets arranged around the periphery of the runner.

CHOICE OF TYPE OF TURBINE

Selection of turbines can be made either based on the available head or on specific speeds :

Choice Based on Available Head :—

The normal selections are given in Table 1.

Table 1.

100 to	1,000	feet	— Reaction
Above	1,000	feet	— Pelton

But in recent years considerable improvement in materials has been made so the reaction wheels have been made upto 1,710 ft. and Kaplan wheels upto 230 ft. as mentioned earlier.

Where the head on the turbine varies considerably from the nominal head, the lower value of the head should be used, and for a small range in heads, not exceeding 10 per cent above or below nominal, the higher heads should be used.

The maximum head under which a Francis turbine operates should not exceed 125 per cent of the design head and the minimum head should not be less than 65 per cent of the design head, in order to obtain stable operation and to avoid excessive cavitation.

Likewise, the maximum head under which a fixed blade propeller wheel operates should not exceed 100 per cent of the design head and the minimum head should not be less than 90 per cent of the design head.

The adjustable blade Kaplan unit may be operated over a head range of 50 to 150 per cent of the design head.

The design head is the head at which the runner is designed for best speed and highest efficiency. Generally, the design head is selected as the head above and below which the average annual generation of power is approximately equal. This selection ensures the most efficient use of the water by having the point of best efficiency at the weighted average head. The rated head is the head at which the full gate output of the turbine will produce the rated capacity of the generator in kW. For run-of-river plants having approximately a constant head, the design and rated heads are usually the same.

Choice Based on Specific Speed

The specific speed of a water turbine is the speed at which the turbine will run when producting one horse power under a head of one foot of water with full guide vane opening.

$N_s = \frac{RPM\sqrt{HP}}{H^{5/4}}$ where H. P. is the output in H. P. and H the head in ft.

in metric unit $N_s = \frac{RPM\sqrt{P}}{H^{5/4}}$ where P is the output in kW and H the head in meters.

The relation between metric and British specific speed is given by N_s (Metric)=N_s (British) $\times 3.8$. If however P is expressed in metric horse power the following relation applies:—

N_s (Metric)=N_s (British) $\times$ 4.45.

The usual choice based on specific speed are given in Table 2.

Table 2

Specific speed	*Type of Turbines*
3-7	Impulse wheel—Single nozzle.
7-12	Impulse wheel—multi nozzle.
19-90	Reaction (Francis)
90-250	Kaplan or Propeller.

Table 3 gives some of the formulae used to arrive at the specific speed for a preliminary design.

Table 3

Francis Turbines:

$N_s = \frac{5050}{H+32}+19$—Crager and Justin (1915-1930)

$= \frac{5050}{H+32}+19$—Barrows (1915-1930)

$= \frac{875}{h\ 0.56}$— Voorduin (h is the critical head and is approximately 90% of the head)

$= \frac{632}{\sqrt{H}}$— (1930-1950)

$= \frac{13000}{H+23}+50$ in metric system (Japanese Electro-Technical Committee).

Impulse Turbines:

$N_s = \frac{175\ H^{3/4}}{\sqrt{HP}}$ for single jet

For multi jet N_s=Speed as calculated above $\times$ No. of jets.

Propeller Turbines:

$N_s = \frac{7000}{H+32}+35$

$= \frac{20000}{H+20}$ in metric system (Japanese Electro-Technical Committee)

The European practice is to adopt specific speeds much higher than the American as can be seen from *Fig.* 3. The use of high specific speed reduces the overall size, weight and cost but requires alower setting to reduce cavitation. With high velocities encountered, they have to resort to a greater use of stainless steel for preventing pitting and wear. Stainless steel is not available in America and it has to be imported.

In the choice of the most suitable speed for the machines for a particular installation and the selection of the type of turbine if the conditions are within the overlapping range of two types—it is important to realise that the highest speed will not necessarily give the lowest overall cost of the installation. An increase of specific speed is usually accompanied by a lower maximum efficiency and a greater depth of excavation necessary for the draft tube; this is particularly so with Kaplan turbines. Thus the choice must be made with full knowledge of the incremental cost of excavation for the foundations and of the value of efficiency. Maximum efficiency may not be the main consideration and weighted efficiency over the anticipated operating range of output is usually the more important criterion.

Experience curve to arrive at preliminary dimensions

An experience curve adopted by the French engineers to arrive at the preliminary dimensions is shown in Fig. 4.

GENERAL DESIGN OF AN IMPULSE WHEEL

(1) In an impulse wheel $N_s \times \frac{D}{d} = a$

constant which lies between 50 & 60.
D—pitch circle dia of runner
and
d—dia of the jet.

With normal and low specific speeds, the buckets can be cast separate and attached by bolts and lugs to the wheel hub. With specific speeds higher than normal the buckets are large and closely spaced and are cast integrally with the hub.

(2) *No of Buckets:* The number of pelton runner buckets for best efficiency could be obtained from the following empirical formula

$Z = 0.5$ m. plus 15

where Z is number of buckets

m=jet ratio$= \frac{D_1}{d_1} = \frac{\text{Mean dia. of bucket wheel}}{\text{least dia. of jet}}$

A normal value for $\frac{D_1}{d_1}$ for efficient operation is about 18. For best efficiency, this is about 10. Any-

thing lower than this will lead to poor efficiency. The D/d, ratio for some of the wheels in service are as follows:—

Pallivasal	10.9
Pykara	13.1
Kundah I	10.3
Kundah II	16.3

The number of buckets is made as large as is judged necessary to give the highest efficiencies and convenient to accommodate on the disc.

Table 4 gives the number of buckets for the impulse wheels installed in Madras State.

Table 4

Power Station.	*H. P. of turbine*	*No. of buckets.*
Pykara	10900 H. P.	26
	18750 H. P.	24
	20000 H. P.	22
Moyar	18000 H. P.	21
Kundah Plant I*	28750 H. P.	22
Kundah Plant II*	50000 H. P.	23

*under construction.

TURBINE CONSTRUCTION

Welded versus cast steel

One of the most important developments in recent years in the design and manufacture of hydraulic turbines is the increased use of welded fabrications for spiral cases, stay rings, head covers, bottom plates, discharge rings and wicket gates. Even Francis runners have been and are being fabricated by welding. The principal reason for this is that while the art of welding has been steadily improving during the past ten to fifteen years, the cost of patterns and steel castings has been increasing relatively faster than the cost of steel plate. Furthermore, it is very difficult to obtain good, sound steel castings, particularly in some of the more intricate shapes. This has led to the steel foundries charging extremely high prices for such items as runner, stay ring and head cover castings.

The large number of repairs required on such castings, together with the delays and costs occasioned by the repairs, have practically forced hydraulic turbine manufacturers to go to fabricated construction.

In welded construction, we may be confident of greater reliability and uniformity, and the possibility of undisclosed flaws is minimised.

By the use of welded plate steel in the place of castings, savings can be made in both the weight of the turbine and cost. For hydraulic turbines, the usual maximum design stress for cast steel is 10,000 lbs./sq. inch for both tension and compression, while for plate steel it is 12,000 lbs./sq. inch. This alone represents a possible saving of 17 per cent in the amount of steel required. Excellent examples of this are the 115,000 H. P. hydraulic turbines supplied for the Hoover Power Plant. The 12 turbines installed in this power plant have cast steel spiral cases. Two additional turbines were installed with cases of welded plate steel. Since seven of the original turbines with cast steel spiral cases and the two recent turbines with plate steel spiral cases were all furnished by one manufacturer, it is possible to make a direct comparison between the two types of construction, both as to weight and cost. Such a comparison shows that the finished welded plate steel casing weighed 30 per cent less than the cast steel casing and cost 20 per cent less to the manufacturer including the cost of the material. Since the spiral casings represent approximately 30 per cent of the total weight of the turbine and 25 per cent of the total cost, the saving in total weight of the turbine was 9 per cent and the saving in total cost 5 per cent.

Plate steel buckets being thinner than cast steel, allow more water to pass through the runner and consequently more power to be obtained for the same size wheel.

The larger savings in both weight and cost plus the other advantages, inherent in a welded steel construction for large turbines with medium and high heads, therefore merit careful consideration.

This also applies to the possible use of field welded joints in place of flanged or riveted joints. During recent years, a large number of the penstocks leading up to turbines have been and are being welded in the field, but field welding of the turbine scroll cases has not been permitted in very many installations.

Some notable exceptions to this are the turbines of the Fontana Project (Tennesses Valley Authority), Shipshaw Plant (Aluminium Co. of Canada) and Cheoah Plant (Aluminium Co. of America) all of which have plate steel casings welded in the field. For the Shipshaw Plant, the plates in the scroll case were welded to each other and to the speed rings at site. No stress relieving or annealing was done. All field welding was done under extenso-meter control. All welds were peened, the amount of peening being controlled by the extenso-meter readings. Specimens were cut daily from the work of each welder and tensile tests were made immediately on a portable testing machine. As a further test of the strength of all the welded joints, the casings were tested under 375 lb. per sq. inch pressure which is double the maximum static head. The units have been in satisfactory service since 1945.

Although these installations demonstrated that welding with mechanical stress relieving is entirely satisfactory, if properly done, some engineers still feel that some form of temperature stress relieving is advisable. Recently, a controlled low temperature method of stress relieving welds in site has been devised, which could be used to stress relieve field welds. Numerous tests have been and are still being made to determine whether this method is satisfactory. No

definite conclusions have been reached as yet, but the results obtained so far are very promising. If this method should prove to be as effective as furnace stress relieving, then the future will see the use of a larger number of spiral casings of all-welded construction.

At Bonneville, for some of the turbines, the stay rings, discharge ring, outer and inner and intermediate head cover, bearing housing pit liner and draft tube liner are all of welded construction. During the war these units were operated with the head water elevation raised by 10 feet above normal, so that the turbines were under about 15 per cent higher head than that for which they were designed. Further, much of the time, the units were running at the cavitation limits. The operating conditions on these large adjustable blade propeller units were such as to place the equipment under severe test.

The welded components comprising the turbine distributor at many plants, both Francis and propeller type, have developed no trouble of any kind.

Introduction of T_1 Steel in Turbine Construction

Four of the largest hydro-electric turbine spiral casings in the U.S.A. of the field welded type using the relatively new T_1 steel at the Noxon Rapids Project. These four Francis type turbines are connected directly to generators of 100,000 kW capacity each, operating under a head 150 ft. The inlet diameter of each spiral casing is 24 ft. and the casings weigh approximately 108 tons each. The casing is designed to withstand maximum design pressure of 85 lbs. per sq. inch without exceeding unit stress of 27,500 lbs. per sq. inch. on any net section with an assumed 90 per cent weld efficiency.

These spiral casings were originally designed for mild steel with maximum thickness of plates equalling 1-5/8″. With the use of T_1 steel the maximum thickness will be reduced to 1″. T_1 steel has excellent properties of notched toughness at low temperature with very high strength and good welding properties. It is a low carbon, quenched and tempered alloy plate steel with a high yield strength of 90,000 lbs. per sq. inch which is approximately 3 times the yield strength of ordinary carbon steel. The tensile strength is 105,000 lbs. per sq. inch.

Table 5 gives the chemical composition of T_1 steel.

Table 5

		percent.
Carbon	..	0.10—0.20
Manganese	..	0.60—1.00
Phosphorous	..	0.040 Max.
Sulphur	..	0.050 Max.
Silicon	..	0.15—0.35
Nickel	..	0.70—1.00
Chromium	..	0.40—0.80
Molybdenum	..	0.40—0.60
Venadium	..	0.03—0.10
Copper	..	0.15—0.50
Boron	..	0.002—0.006

Table 6 gives the mechanical properties of T_1 steel for thickness 1/4″ to 2″.

Table 6

Yield strength (Min.)	..	90,000 psi
Tensile strength (Min.)	..	105,000/135,000 psi
Elongation in 2″ (Min.)	..	18%
Reduction of Area (Min.)	..	55%

T_1 steel (thickness for thickness) costs more as the material is more difficult to fabricate, handle and weld.

Table No. 7 gives the approximation of the comparable cost factors between carbon steel and T_1 steel steel based on experience. One comparison is between carbon steel and T_1 steel of the same thickness. The second comparison is between carbon steel and T_1 steel which is 44% of the thickness of the carbon steel which represents the ratio between 12,000 lbs. per sq. inch designed stress for carbon steel and 25,000 lbs. per sq. inch for T_1 steel.

Table 7

	Carbon steel	*same thickness as carbon steel*	*44% of thickness of carbon steel*
Plate material.	A	2.5 A	1.1 A
itting.	B	2.0 B	.88 B
Fabrication.	C	1.4 C	.62 C
Erection.	D	1.1 D	.48 D
Welding.	E	1.5 E	.66 E
Freight.	G	G	.44 G

Table 7 indicates there is a saving in use of T-1 steel in all factors except the cost of the materials, provided full advantage can be taken of its higher strength. These cost factors are based upon rather meagre information and it is quite possible that the costs involved in T-1 steel will be reduced after more experience has been obtained in its use and application.

It is also probable that the design stress for this steel can be increased to make its use still more advantageous.

T-1 steel welds do not require stress relieving and there is no advantage obtained by such stress relieving,

Table No. 8 indicates some of the units where T-1 steel have been used for spiral casings.

Table 8

Name of Plant	*No. of units*	*Design Press in p. s. i.*	*Hd. ft.*	*H.P. capacity*	*Inlet dia.*	*Max. Min. plate thick*	*Field Press Test*
Idaho Power Co. Brownlee Plant	4	185	250	158000	18′-0″	15/18″	No
Pacific Power & Light swift Creek No. 1.	3	205	378	117000	11′-0″	13/16″-1/2″	No
Washington Water Power Co., Noxon Rapids.	4	85	152	137500	24′-0″	3/4″ 1-2″	128

The minimum thickness recommended for T-1 plates is 1/2″ or in some cases 3/8″. T-1 steel would therefore not be suitable for medium sized hydraulic turbine units for low head installations where the maximum plate thickness for carbon steel will be only 1″ or less because full advantage could not taken of the possible reduction in thickness due to the minimum thickness limitations of the T—1 steel. Thus, the economic savings would be reduced or eliminated entirely.

Runner

Materials :—The choice of materials varies considerably from one manufacturer to another. In certain countries, where good castings, are difficult to obtain locally in a reasonable time, there is a marked trend towards welded construction. On the whole, more European manufacturers still make a liberal use of cast iron and cast-steel. These materials, though heavier than fabricated steel, are not necessarily more costly and they have the advantage, for large machines particularly, of being more rigid.

The situation is similar as regards the use of stainless steel. Castings are used liberally whenever good ones are obtainable at reasonable prices; the use of stainless steel over-layers, in strips or by welding, is resorted to mainly in those cases where solid cast or forged stainless steel elements are not readily obtainable at reasonable prices.

Attempts have been made to fabricate snch complicated parts as Francis runners or Pelton buckets with cast and laminated stainless steel elements welded together, but these still are the exception, and the present trend is definitely in the opposite direction, i.e. towards integrally cast runners.

In the case of Pelton wheels, for example, in the majority of cases all the runner buckets are cast in one piece, either in the form of a ring of buckets or integrally with the disc and coupling flange, whether the runners are made of carbon steel, nickel steel, or chromium steel; only in exceptional cases do certain European manufacturers still resort occasionally to bolted bucket construction.

Stainless Steel:—In addition to the increased use of fabricated plate steel and welding in hydraulic turbine construction, there is a tendency towards the increased use of stainless steel. The stainless steel has the property of not only resisting corrosive action due to chemicals in the water, but also resists the pitting action due to cavitation.

There have been cases where a combination of the corrosive action of the water, high water temperature, and cavitation, produce an extremely rapid rate of wear not only on the runner but on the wicket gates, wicket gate stems, head cover, and bottom plate. This is eliminated by using a solid stainless steel runner. The corrosive action on the wicket gates, wicket gate stems, head cover and bottom plate was eliminated by the use of stainless steel facings and stainless steel wearing plates.

Solid stainless steel Francis runners have been used for a number of high head installations and the use of such runners are recommended for heads above 600-700 feet. It is also desirable to use stainless steel wicket gates, facing plates and wearing rings for high head installations.

Operating and maintenance experience with high head Francis turbines has indicated that the additional initial cost of using stainless steel for these parts is well repaid in reducing maintaenance cost and outage.

The grade of stainless steel which in Europe is most favoured for water turbine construction is that containing 12 to 14 per cent chrome with or without a touch of nickel (approximately 1 per cent). These alloys have a high resistance to cavitation and can be treated to give a mechanical resistance which is considerably higher than that obtainable with the 18-8 grade of stainless steel. They, therefore, resist erosion better and are at the same time easier to machine and less liable to seizure. Moreover, they can be welded in the field, provided that certain precautions are taken (pre-heating of the parent metal) without its being necessary to heat treat the welded metal. However, 18-8 grades of steel in this respect are more favourable than the 13-1 grades.

Bronze:—The fact that they are weldable is one of the main reasons for which stainless steel in many cases are preferred to manganese or phospor bronze

for turbine parts subject to cavitation. However, bronze is still much in favour because, in actual practice, it resists pitting remarkably well even though certain laboratory tests class bronze below stainless steel in the range of corrosion resisting alloys. This is doubtless owing to the fact that the experimental devices used in these research laboratories for accelerating the pitting of test bars give rise to a form of cavitation which is not entirely comparable with that which occurs in a Francis turbine, for example.

Among the many bronze runners which have given good service are those of the 32,0000 h.p. Piottino Francis turbines in Switzerland operating under a head of over 1,000 ft.

Ampco Bronze:—Another source of damage to hydraulic turbines which frequently combines with cavitation and corrosion is erosion caused by foreign matter such as silt or sand or glacial deposits in the water. When water at high velocity carries such hard materials in suspension, the damage can be rapid and severe.

Unfortunately, both research tests and actual field operation have indicated that the 18-8 stainless steel which have a high resistance to pitting, do not have a very high resistance to erosion. In this respect, the chrome stainless steel with 12-14 per cent chrome content show much better resistance than 18-8 chrome-nickel stainless steel. Other promising metals for resistance to erosion are the aluminium bronze, particularly those that are produced under the trade name of Ampco Bronze. Ampco is the trade name for a bronze with an aluminium content varying from 10 per cent to approximately 14 per cent. The results of tests on the cast bronzes show that some of them have twice the resistance to pitting compared to the best stainless steel castings. The welded bronzes also show a remarkable resistance to pitting. In the HEPC Ontario, Ampco bronze runners are installed in units which are subject to large quantities of highly abrasive sand in the water. The original manganese-bronze runners lasted only a short time and after several extensive repairs had to be replaced once every five years. 18-8 stainless steel runners were installed to replace two of the manganesebronze runners. The runners have stood up better than the manganese-bronze runners in that they showed no signs of pitting. They do show however evidence of sand erosion, although considerably less than obtained in manganese-bronze runners during a similar period of time. Ampco bronze runners were installed to replace the last two manganese-bronze runners. These have stood remarkably well during their period of operation, showing no sign of either cavitation or erosion. The manganese steel runners at Pykara have also shown signs of pitting and wear much earlier than stainless steel runners.

Accelerated cavitation test on Ampco bronze have shown that it is highly resistant to pitting. However, its one big disadvantage is that at present it cannot be welded overhead. If an ampco bronze runner requires welding repairs it must be removed from the turbine, since pitting usually occurs on the bottom side of the runner.

Wicket Gates

The wicket gates or guide vanes as they are sometimes called, are arranged in series in a circle surrounding the runner and may vary in number from 12 to 24 depending upon the physical size of the unit. They are of such cross sectional shape as to fit into the stream flow at all loads with the minimum of disturbance. Their function is to control the flow of water to meet the power demand. They contact each other when fully closed. They are machined at the contact points and fitted at these points to a .003" feeler.

The wicket gates are made usually of cast steel with stems cast integrally. For low head installations turbine gates of fabricated plate steel with forced steel stems are acceptable. Three bronze bushed stem bearings are required one in the bottom ring and two in the head cover, one above and one below the packing box. Positive means are provided for lubricating each bearing independently.

The largest items of maintenance cost on Francis turbines next to maintenance of the runner are the wicket gate stems for both low and high head installations. The wear on these parts is due to corrosive action of the water. The use of stainless steel sleeves for the stems tends to reduce such wear to a minimum and such sleeves have been installed in a number of recent installations. Wear along the discharge ends of the wicket gates and along the seal point presents a problem in medium head and high head installations. The insertion of stainless steel strips at these points tends to reduce this wear considerably. Undercutting of the wicket gates where they form a seal with the head cover and bottom plate is also a common occurence in the higher head installations. This is overcome by using stainless steel plates or welded overlays on the ends of the gates. The use of stainless steel for these parts is justified in large turbines where outage for repairs is costly and where an operation life of 20 to 30 years is now expected without a major overhaul of the turbine.

Inlaid rubber strips have also been used on the gate seals and in the head covers and bottom plates to reduce leakage and thus reduce wire-drawing and undercutting. These rubber strips have given satisfactory service on a number of turbines. The turbine gates may be locked by the following method to prevent excess load on the generator. A mechanical block on one of the guide vane cylinders can be adjusted to limit the maximum guide vane opening and the governor is equipped with a gate limit device which limits the permissible gate opening.

Since it is possible for a foreign object such as a log, to get between two wicket gates in service and for the whole of the servo motor capacity to be concentrated on two vanes, it is usual to incorporate into the design of the linkage connecting the wicket gates to the gate operating ring a safety feature such that the stress in the stem will not exceed the yield point of the materials.

AIR VALVE

When a unit is operating below rated power, the water passages inside the wicket gates may not be full of water. This may result in a periodic breaking of the water column which can in turn be the cause of severe shocks and vibration in the power house structure. The admission of air under these conditions reduces the vacuum and tends to stop the breaking of the water column, resulting in a smoother operating unit at part load, one or more air valves depending upon the size of the unit and the amount of air considered necessary, are usually fitted and controlled from the operating mechanism.

DRAFT TUBE

The draft tube is an extension of the passage from the point of exit from wheel runner down to tail race level. This forms an internal part of the turbine in determining the turbine performance.

The draft tube supplements the action of the runner by utilising most of the energy remaining in the water at the discharge from the runner. This is shaped to decelerate the flow with a minimum of losses so that the kinetic energy remaining in the flow at discharge from the runner may be efficiently regained by conversion to the suction head. thereby increasing the total pressure difference on the runner.

Preliminary Dimensions

The preliminary dimensions of the draft tube are fixed by the wheel dimensions using the following empirical formulae. Fig. 5.

(*i*) Maximum width of draft tube. – 3 D_2
(*ii*) Vertical length from the centre line of the distributor to the bottom of elbow draft tube. .. .. –2.7 D_2
(*iii*) Horizontal length of the turbine shaft centre line to discharge end of of the elbow draft tube. .. –3.8 D_2

(*iv*) $\frac{D_2^3 \times n}{Q}$ lies between 85 and 115

Where Q is the discharges in cusecs and n speed in RPM D_2—dia of runner in ft.

Table 9 gives the draft tube details of some plants in operation.

Table 9

	Discharge cusecs	*Head*	*Mean velocity at draft tube ft. per second* *Draft Head*	*Entrance*	*Exit*	*Gauge pressure PSI inches.*
1	2	3	4	5	6	7
Leaburg	1014	89.0	6.5	16.0	6.8	—4.0
Safe Harbour	8000	55.0	1.0	26.7	8.1	—4.4
Oak Grove	400	850.0	10.0	15.1	3.7	—5.3
Norris	4100	165.0	11.5	27.3	6.2	—9.3
Bonneville	12000	45.0	—7.0	29.0	5.1	—3.6
Bonneville	10000	64.0	—1.4	24.2	4.3	—1.9

It will be observed from the table that for Bonneville, under a head of 45 ft. $\left(\frac{29 \times 29}{64} \text{ or } 13.2 \text{ ft. i. e. } \frac{13.2}{45} \right)$ or nearly 30% of the energy remains in the water after it leaves the runner ; which emphasises the need for careful draft tube design in order to realise as much of this as possible.

Velocities

The following are the empirical formulae for the velocity of water in draft tubes.

(*i*) At the entrance to scroll case.
- $0.13 \sqrt{2gH}$ for high head plants.
- $0.07 \sqrt{2gH}$ for low head plants.

(*ii*) At the exit of the runner :
- $0.4 \sqrt{2gH}$ for low head plants.
- $0.1 \sqrt{2gH}$ for high head plants.

(*iii*) At the outlet of the draft tube:
- $0.10 \sqrt{2gH}$ for low head plants.
- $0.02 \sqrt{2gH}$ for high head plants.

Submergence

It is important that careful attention is paid to the amount of submergence at draft tube outlet to avoid undue turbulence and in some cases an actual evacuating effects with consequent tail water lowering at the draft tube discharge.

For small high head units, it may be necessary on this account, to increase the proportions form the top of the draft tube to the bottom thereof at discharge outlet to obtain amples submergence. For such units we should have a submergence, from the upper edge of the draft tube discharge to low tail water of not less than 3 feet, and when the head is high, say about 600 ft., it will be well to increase this to a minimum of of 4 feet. In no case, should this distance be less than 2 feet regardless of the working head.

Generally this should not be less than one half runner diameter and the lowest tail water level should not uncover any part of the roof of the horizontal leg of the draft tube.

The draft tube when not designed by the turbine manufacturer should follow the standard elbow design so far as practicable, but certain variations in the lengths of the vertical section and horizontal diffuser are permissible. The outflow velocity may vary from 4 to 8 feet/sec. with good efficiency.

A double pier draft tube is preferred for hydraulic reasons, but for practical reasons a single pier draft tube may be used on runners of from 4 to 7 ft. discharge diameter.

SPEED AND PRESSURE REGULATION

General

The ability to predetermine the speed changes on a hydraulic turbine in terms of the maximum change in speed or the variation in speed with respect to time, permits the complete study of regulation problems in advance of the actual construction of a hydro-electric project. It is often desirable to know the amount of speed change occurring at various times following a load change in order to make adjustment of the electrical protective devices or to determine the possibility of over voltage conditions in the stator of the generator or on transmission systems due to sudden load changes.

The normal maximum pressure rise admitted in the cases of Francis turbines is of the order of 30 per cent but in several cases a pressure rise of 40 per cent or even more has been permitted in French practice.

In the case of Pelton wheels, however, a maximum presure rise of between 10 per cent and 20 per cent is normally adopted.

With the length, diameter and profile of the penstock predetermined, considering, in the case of high heads a long water conduit with a properly designed surge tank or open forebay constitutes a point of relief as defined in the study of water hammer, then the minimum time of closure of the turbine gates depends upon the maximum rise is pressure allowable for the strength of the penstock or upon the maximum fall in pressure permissible to avoid formation of vacuum in the penstock due to acceleration of flow.

It is usual to design hydro-electric plants with the maximum rise in pressure not in excess of 50 per cent of the normal head on the turbine, but it is frequently found desirable to limit the maximum rise to 25 per cent of the normal head or even less to avoid excessive stresses in the penstock which may be repeated from time to time due to the changes in load on the unit. Recent experiments have indicated that excessive water hammer occurring at frequent intervals may cause fatigue of the materials in the penstock and increase the danger of rupture following water hammer surges.

In certain installations, the rise in pressure is taken care of by providing an auxiliary relief valve opened by the governor as the gates are closed, thus by-passing the water and reducing the rate of change in velocity of the water to a minimum. In calculating the speed rise for such cases, the quantity of water discharged by the relief valve is deducted from the total flow in computing the pressure rise in pipe and change in velocity.

In the case of on-coming loads the profile of the penstock should be considered and the maximum fall in pressure calculated, the acceleration gradient determined an the minimum allowable opening stroke of the governor established in this manner.

In determining the flow through the conduit the rated capacity of the unit should be taken and the discharge corresponding to this output at efficiency of 85 to 88 per cent calculated. Although the actual efficiency may be greater than this there is some overload capacity allowed behind the rating and the use of the lower efficiency will compensate for this as an added margin of safety.

The permissible governor time may be based on the load change from 50 per cent to zero without breaking the draft tube vacuum, since the condition for full load thrown off is sometimes too severe and results in too long a governor time, although in certain cases it may be justifiable to utilize the full load condition of design.

In case the method of computing governor time show a permissible rate of movement of the turbine gates for a full load change of less than 2 or 3 seconds, it is advisable to limit the governor motion to a slower rate of closure for purely practical reasons. With large units having a rapid governor stroke the governor itself must be quite large an the piping leading from the governor to the operating cylinders must be made oversize to avoid high velocities of the governor fluid resulting from a full stroke in a very short time. It is rarely necessary to go below the governor time of 3 seconds in order to meet ordinary commercial regulation practice in regard to speed variation and, if a governor time of 3 seconds is utilised for plants having appreciable length of penstocks even under medium head it will be found that more satisfactory hyraulic-conditions will result than from the use of a much shorter time. In high head plants it may be desirable to go as high as 4 or 5 seconds for a full stroke and possibly more than this in certain cases.

In all calculation of WR^2, governor time, etc., it is assumed that the unit is isolated from the system without securing any benefit of connected fly-wheel effect of other generating units or of the load which it is carrying. In actual practice for small load changes which do not change the speed sufficiently to trip the relay devices, the connected flywheel effect of the system assists materially in reducing the speed fluctuations.

Pressure Regulation

For all types of plants, even those operated under high heads with long pipe line, it is usual to adopt a closing or opening time of the guide vanes which is much greater than the time of reflection of the pressure wave. With normal operation, i.e., with loads thrown on and off, either slowly as the system requires or rapildly during faults, the pressure regulations of water turbines fall almost antirely within the category of slow surges.

Fig. 6 gives the change of pressure for any given set of conditions of water quantity and waterway dimensions. The momentum and starting time of the waterways in the calculation are defined as given below:

Momentum waterway $= LV$

Starting time of waterways (hydraulic inertia) $\Big\}$ $\dfrac{LV}{gh}$ in secs.

where L = Length of the pipe line

V = Velocity of water in ft./sec.

H = Working head.

The pressure change is obtained from the figure after calculating the value $\frac{\Theta}{T_e}$

Where T_e — governor closing or opening time.

When pressure rise is considered, net head should be used for calculating Θ and friction losses in the pipe line added to the calculated pressure rise to obtain the maximum pressure. In the same way for pressure drop, initial head is used for calculating Θ and friction losses subtracted to arrive at the lowest pressure.

Speed and Pressure Rise and Speed Drop of Units in Madras State

The pressure and speed rise for full load thrown off condtiions of the turbines in operation in the Madras Grid are given in table 10.

Table 10

Plant		*Pressure Rise*	*Speed Rise*
Pelton:			
1. Pykara	7810 kVA	11.5%	13%
	12500 kVA	10%	11%
	16000 kVA	10%	10%
2. Moyar	12000 kW	10%	10%
3. Kundah (i)	24000 kVA	15%	30%
(ii)	42000 kVA	10%	30%
Francis:			
1. Mettur	10000 kW	20%	30%
2. Papanasam	7000 kW	10%	13.5%
3. Machkund (Andhra)	17000 kW	15%	16.5%
4. Periyar	35000 kW	30%	11%

Table 11 gives the speed drop for some of the units referred in Table 10.

Table 11

Station		*Speed drop%*
Pelton		
Pykara	7810 kVA	60
Moyar	12000 kW	25
Kundah — 1	24000 kVA	30
Kundah — 2	42000 kVA	30
Francis		
Mettur	10000 kW	18
Papanasam	7000 kW	22.5
Machkund (Andhra)	17000 kW	25 (50% suddenly)
Periyar	35000 kW	30

Pressure Regulators

Pressure regulators may be desirable in high head installations to permit rapid closing of the turbine wicket gates without creating excessive pressure rise.

They are ordinarily not required unless the length of the penstock from the forebay or surge tank to the turbines exceeds 5 times the maximum net effective head.

Pressure regulators may be water saving or water wasting type. A water wasting pressure regulator is designed to maintain constant velocity in the pipe line at all conditions of load. The opening of the pressure regulator is adjusted in proportion to the gate opening so that water not used by the turbine will be discharged through the pressure regulator. This system does not allow economical use of water, but may be necessary where there are rapid load fluctuations especially where there may be instantaneous demand for a large increase in power.

Water saving pressure regulators are designed to open only when the turbine gates are closed rapidly and are designed to close slowly so that there will be no serious over pressure set up in the penstock. Many pressure regulators are now constructed so that they may be operated either as water saving or water wasting type.

Where plants have relatively short penstock lines or they can operate on block load with governors under the control of load limiting devices, conditions are ideal for the use of water saving relief valves. In this case, the relief valve is arranged to open practically synchronously with the closure of the turbines gates and then to close gradually and stop the flow of water through the penstock.

The size of capacity of the relief valve has a direct relation to the length of penstock and an inverse relation to the time of governor stroke.

On a very short penstock with a low head, the relief valve may be omitted with safety, but with a long penstock it is necessary to have a relief valve capable of passing the maximum flow of water taken by the turbine.

Between these limits, it is customary practice to use relief valves having from 50 to 80 per cent capacity provided that the pressure rise due to turbine gate closure does not exceed 20 to 25 per cent of static head. There is an approximate criterion that states that when the time of governor action is shorter than $\frac{\text{Length of penstock} \times \text{Velocity in penstock}}{10 \times \text{Head}}$ a relief valve should be installed. This is only an approximate rule, however, and the particular problem should always be thoroughly investigated, balancing the cost of the relief valve against the cost of the penstock and hydraulic equipment necessary to withstand the increased pressure which would result if the relief valve is omitted.

Pressure regulators should be avoided unless they can be justified from an operating stand point as for example when need arises for a synchronous by-pass to maintain a constant flow in long canals or tunnels or penstocks where the flow cannot be readily stopped or started or when a unit at the end of a long penstock is required to pick up or reject a load suddenly.

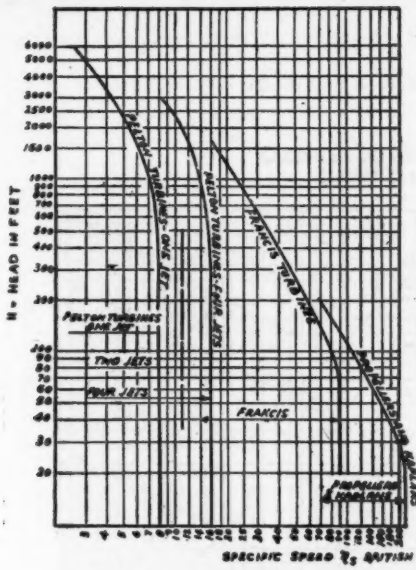


FIG. 1.

LIMITS OF HEAD FOR VARIOUS SPECIFIC SPEEDS

THE FIELD OF OPERATION FOR EACH TYPE OF TURBINE IS LIMITED BY ITS HIGHEST AND LOWEST N_S AND BY THE MAXIMUM HEAD UNDER WHICH IT CAN SAFELY BE OPERATED

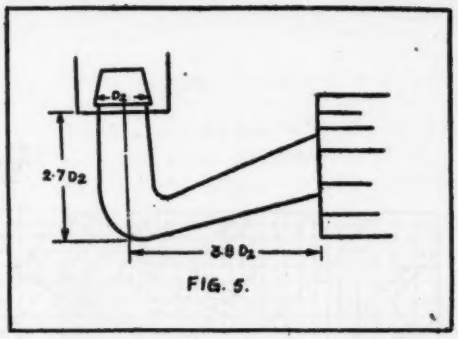


FIG. 5.

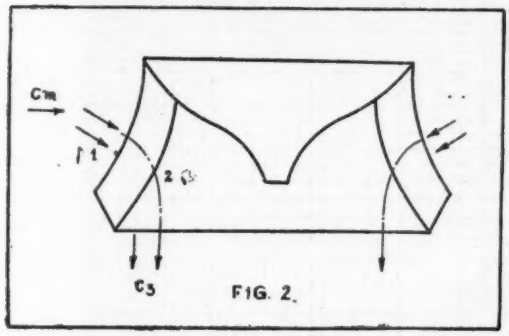
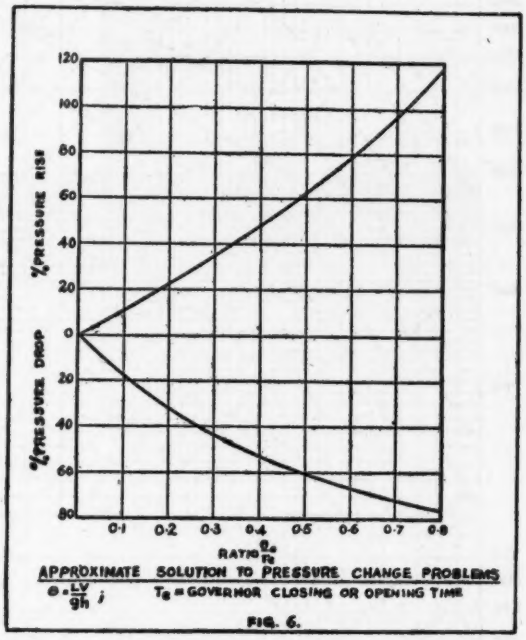


FIG. 2.



APPROXIMATE SOLUTION TO PRESSURE CHANGE PROBLEMS

$\theta = \frac{LV}{gh}\,i$ $\quad T_s$ = GOVERNOR CLOSING OR OPENING TIME

FIG. 6.

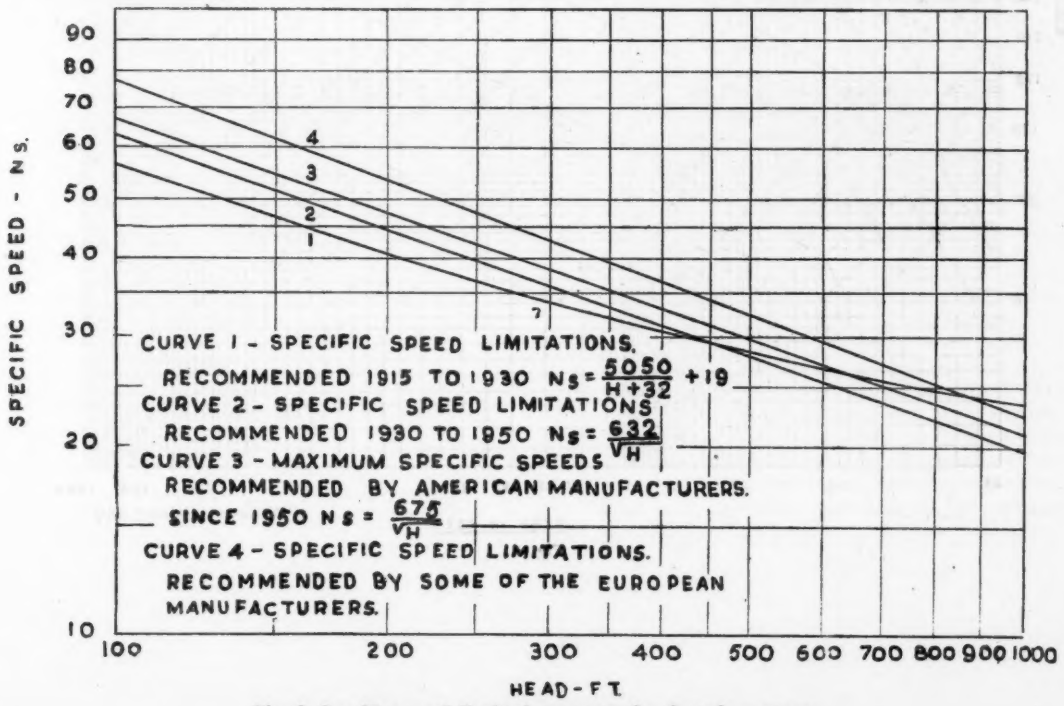

SPECIFIC SPEED - Ns.
HEAD - FT.
CURVE 1 - SPECIFIC SPEED LIMITATIONS.
RECOMMENDED 1915 TO 1930 $N_S = \frac{5050}{H+32} + 19$
CURVE 2 - SPECIFIC SPEED LIMITATIONS
RECOMMENDED 1930 TO 1950 $N_S = \frac{632}{\sqrt{H}}$
CURVE 3 - MAXIMUM SPECIFIC SPEEDS
RECOMMENDED BY AMERICAN MANUFACTURERS.
SINCE 1950 $N_S = \frac{675}{\sqrt{H}}$
CURVE 4 - SPECIFIC SPEED LIMITATIONS.
RECOMMENDED BY SOME OF THE EUROPEAN MANUFACTURERS.

Fig. 3. Specific speed limitation curves for francis runners.

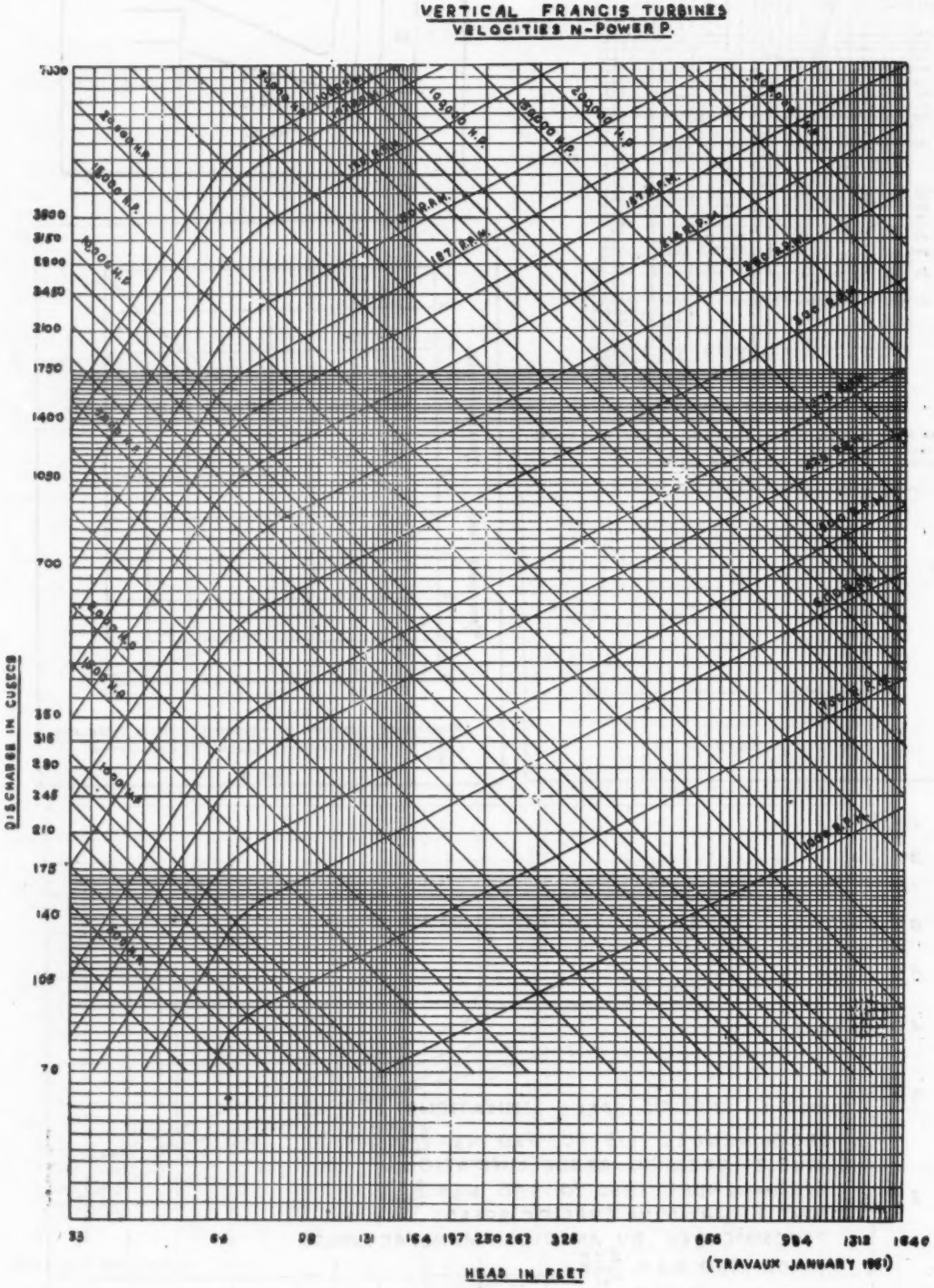

VERTICAL FRANCIS TURBINES
VELOCITIES N-POWER P.
DISCHARGE IN CUSECS
HEAD IN FEET
(TRAVAUX JANUARY 1951)

Fig. 4.

CAVITATION IN WATER WHEELS

Cavitation is defined as the formation of voids within a body of moving liquid, when the particles of the liquid fail to adhere to the boundaries of the passage way. The failure of the particles to adhere to the boundaries occurs when there is insufficient internal pressure to overcome the inertia of the particles and force them to the sufficiently curved parts along the boundary. The voids or cavities thus formed become filled with the vapour of the liquid and cause formation of vapour bubbles. Since the inertia of a particle of liquid varies as the square of the velocity and since greater the inertia, the greater the pressure required to force the particle to take a curved path. it is apparent why cavitation is usually associated with high velocities and low pressures.

Cavitation occurs in hydraulic turbines usually on the underside of the runners and the top of the draft tube where the pressures are below atmospheric to account of the elevation of the turbine runner over the tail water.

The turbines should be set with respect to tail water at the economic elevation based on structural requirements provided that this results in a setting low enough that the maximum permissible total draft head or suction pressure on the discharge side of the runner will not be exceeded. Draft heads in excess of the maximum permissible value may produce cavitation which will cause a reduction in efficiency and will result in gradual destruction of the metal in the runner.

For a given runner and draft tube the upper limit of the suction head (H_s) is determined by the occurrence of cavitation phenomenon and the extent of the suction head can be computed from an empirical formula or from various equations deduced by various authorities.

The figures thus obtained are quite reliable in case of small units operating at normal heads but model tests must be resorted to when dealing with large high head turbines.

The elevation finally selected for the runner should ensure a pressure head that will result in undisturbed running and safeguard against cavitation as well as against turbulence in the draft tube during governing operation.

The maximum permissible total draft head H_s is determined by the formula

(*i*) For francis wheels

$$H = \frac{H_b - H_s}{0.625 \left(\frac{N_s}{100}\right)^2}$$

(*ii*) For propeller wheels

$$H = \frac{H_b - H_s}{0.25 + \frac{1}{7.5} \times \left(\frac{N_s}{100}\right)^2}$$

Wnere H_b is the atmospheric head minus vapour pressure head

H is the operating head and H_s is the suction head

In a water wheel operation

$$HM = H_a - H_s - \sigma H$$

$$\text{or } \sigma = \frac{H_a - H_m - H_s}{H}$$

Where

σ = is the cavitation factor. This is the value at which cavitation begins. In an actual installation should be higher than the critical value.

H_m = is the head in feet of vapour pressure.

H_a = is the barometric pressure is feet of water column.

H_s = is the suction head.

H = is the actual head acting upon the turbine, measured at the centre line of the runner.

In a given installation, if the head water and tail water installations are simultaneously reduced by equal amounts, keeping the effective head the same, H_s will be inreased and H_m correspondingly reduced. If this reduction of H_m is carried far enough, it will reach a limit, below which it cannot go, (i.e.) below the vapour pressure of water. When it reaches this point the water at the local point in the stream where H_m occurs, will begin to vaporise and form vapour filled cavities within the flowing body of liquid.

At this point, the coherence of the water is broken and air with its constituent elements 0_2 and N_2 and etc., is freed from the water and a series of bubbles are formed which further aid in the vapourising of the water. These small voids or bubbles occur in eddies or vortices. As these pass over the casing of the runner, they get into higher pressure regions and suddenly collapse.

The formation of these pockets or cavities is known as cavitation. The formation of cavitation at vapour pressure conditions is common experience and it implies that water has no tensile strength, while the breaking strength of water between its molecular layers has been estimated at 150,000 p.s.i. This condition where cavitation begins is a critical point in the operation of the turbine.

If a turbine is installed with the runner too high above tail water or sump level so that the critical point of cavitation is reached, the water then flows to the region of minimum pressure and vapourises. As it passes on, beyond this local point to a region where the pressure is above the vapour pressure, the water will then return from the vapour state to the liquid. It will suddenly condense. Moreover the pressure at a given point in such a stream is never perfectly steady and vapourisation and subsequent condensation can occur at the same spot. The effect of the rapid condensation and the collapse of the cavity, is of the nature of an explosion and repeated reforming and collapsing of the cavities sometimes sounds like a machine gun. Serious mechanical vibration and unstable operation of the machine may develop.

These objectionable effects of cavitation like impairment of efficiency and power, vibration and instability of operation are, however, overshadowed in importance by the most serious consequences of cavitation, viz., pitting of the metal surfaces exposed to it.

To avoid this plague which sometimes sets in, the only safe procedure is to determine the cavitation coefficient of any machine which is to be installed, and to see that the static draft head required by the setting is not excessive for the particular design of the machine. For low speed machines, the cavitation characteristics may be usually estimated on the basis of experience with the installations of similar type, but with high speed type turbines such as propeller turbines, the only safe procedure is to make a cavitation test in the Laboratory of an exact model of each contemplated installation, covering a range of conditions equivalent to the full range of heads and loads to be encountered in the field.

There are many theories concerning the actual mechanics of the pitting action, but as yet none of them has been proved to the point where it is accepted as actual fact. One of the most popular theories is that the extreme localised pressure developed upon collapse of the vapour bubbles forces the water into the pores of microscopic cracks in the metal or between the grains in the metal structure where the water is actually compressed. As the pressure is relieved, perhaps due to the formation of another vapour bubble the water expands and actually explodes in the metal tearing away the surface metal at that localised point and causing pitting.

Pitting is differentiated from cavitation as to actual erosion of the material under the pressures produced by the collapse of bubbles formed by cavitation. Experiments on the actual formation, growth and collapse of bubbles indicated that the bubbles were about $\frac{1}{4}$ inch in diameter, their life was .003 seconds and their maximum velocity of collapse was about 765 feet per second, occurring at the instant of complete collapse and the force of impact has been estimated to reach 100000 tons per sq. inch.

Actual tests on a venturi with a $\frac{1}{4}''$ diameter throat showed that cavitation started with a velocity of flow of about 55 feet per second and absolute pressure of 0.5 psi. in the throat.

From the water hammer formula $P = 65\,V$, it can be calculated that the collapse of each bubble produces a pressure of at least 50,000 psi. $(P\text{-}65 \times 765) = 50{,}000$ psi.

This pressure is of course concentrated upon a microscopically small area which enables it to attack the individual grains or even the grain, boundaries in material exposed to cavitation. Since the bubbles form and collapse at the rate of 10,000 to 20,000 per second it is apparent that the destruction of the metal is largely due to a fatigue failure.

Cavitation can be reduced or eliminated only by increasing the back pressure from the tail race or the discharge side of the turbine blades or by decreasing the velocity of flow. The pressure on the discharge side can only be increased by setting the turbine runner low with respect to tail water. The velocity can be reduced by increasing the size of the runner.

The surfaces of a hydraulic turbine which are normally affected by the action of cavitation are the discharge surfaces of the buckets, or blades areas on the crown, inside surfaces of the shroud band, on the throat liner, at the tips of the blades, and the upper portion of the draft tube. In rare cases there may be pitting on the pressure faces of the buckets or blades due to an unusual amount of overhang of the wicket gates, improper design or unusual operating conditions. Pitting of the outside of the shroud band may also occur if hydraulic conditions in the draft tube are such as to develop considerable vacuum in that region. Pitting generally occurs on the intake sides of the blades or vanes, of the pump impeller, the casing, near the tips of the vanes, or other low pressure areas. Cavitation in hydraulic turbines usually affects the performance adversely from the stand point of efficiency and power and the life of the tubines parts. It also makes the production of power and efficiency unreliable. Cavitation damage is probably responsible for more time out for repairs than all other mechanical difficulties.

Safeguards against Cavitation

(*a*) The most effective and surest way to prevent cavitation is to set the runner comparatively low with respect to the tail water elevation by making H_s small or even negative, high value for power plant sigma will be obtained. The higher the value for power plant sigma compared to the cavitation sigma of the runner, the less possibility there is of cavitation and consequent pitting. This simply means that a high back pressure is built up on the discharge side of the runner which prevents the formation of cavitation voids. However, it is not always desirable from an economical stand point to set a runner sufficiently low to eliminate all cavitation, since this may result in excessive excavation and foundation costs.

(*b*) Another method of reducing or eliminating cavitation is to reduce discharge velocities of the runner. The lower the velocity the less tendency there is for cavitation. However, this calls for an increase in the size of the turbines for the same output.

(*c*) The shape of the turbine runner blades will also effect formation of cavitation. Sharp or sudden change in shape is a source of cavitation, because of the inability of water particles to follow the changes in shape under certain conditions of velocity and pressure. This charateristic is taken into account by the designer who attempts to so shape the runner blades as to make flow passages smooth and eliminate all abrupt changes, humps and depressions.

There is a tendency today to use higher turbine speeds and to increase the power from a given size of machine. This is due to a general improvement in hydraulic designs and to a better understanding of what can be done without risk of serious cavitation damage. Pushing too far in this direction without adequate information can be a serious mistake; on the other hand, using too conservative a design because of

lack of information is also wrong because unnecessary capital charges on equipment are as real as repair bills.

Much can be done to prevent cavitation damage by careful design of the hydraulic turbine especially of the runner. Every effort is made to reduce the tendency to cavitate by preventing sudden changes of the direction of water flow in low pressure areas and by forming the surfaces in these regions smoothly.

Pitting of turbine runners is known to occur in all types of runners and kinds of metals. In atmospheric corrosion, the products of corrosion remain on the surface, while with pitting of the runner, the products are carried away and the pitting surface exposed to further attack. In severe cases runner buskets 2" thick have been eaten through in less than 2 years.

Nearly all the pitting of turbine runners has occurred on the back of vacuum side of the wheel, suggesting that the degree of vacuum or the draft head acting on the runner is a factor in pitting.

The action due to cavitation is therefore fundamentally mechanical.

From an operating standpoint, higher speed, higher head, or greater capacity, lowering of tail water or improving of draft tube efficiency and finally especially under high heads operating at gate openings other than the best gate opening all contribute towards pitting. Runner design as to bucket size, angle of curvature as affecting tendency of the stream to pull away, draft tube design, all influence pitting.

Pitting Resistance of various Runner Metals

All metals presently used in the manufacture of hydraulic machinery are affected and damaged in varying degrees by the action of cavitation. Stainless steel and Ampco bronze either cast or deposited by welding are nearly impervious to cavitations. These metals when applied over areas subject to cavitation are called anti-cavitation coatings.

Table 12 gives the relatives loss of metals due to cavitation.

Table 12

PITTING RESSISTANCE OF VARIOUS TURBINE RUNNER METALS

Type of Material	*Relative rate of Pitting*
Welded stainless steel 17 Cr—7 Ni	1
Cast stainless steel 12% Cr	3
Welded stainless steel 18 Cr-8 Ni	5
Cast stainless steel 18-8	5
Aluminium bronze	25
Cast steel 0.33 C	37
Manganese Bronze	80
Sprayed stainless steel 18-8	122
Cast iron	224-375

COMPARISON OF PELTON AND FRANCIS WHEELS

Head

A pelton wheel must always be set at a higher level than the highest tail-race level and therefore shows a loss of head when compared with the Francis turbine which due to the draft-tube makes use of the total available head.

The Francis turbine therefore shows a distinct advantage, particularly in installations in which the variations of the tail race level are important. In the case, for example of a plant-operating with a head of 1000 ft. and having a tail race level varying by 10 ft. the Francis turbine which can operate perfectly well with a 'back pressure' can make full use of the total head even if the conditions to be observed to avoid cavitation require that the runner should be placed at a level very near or below the lowest tail-race level. In the case of the Pelton, however, the runner will necessarily have to be placed at least 6 ft. above the highest tail-race level, and hence in respect of the operating head sustains permanent loss of between 0.6 and 1.6 per cent.

A further advantage of the Francis wheel lies in the fact that it accommodates variations of operating head far better than a Pelton wheel.

Francis turbines adjust themselves be ter to varying water levels in a storage reservoir and will therefore use this storage to better advantage and with an improved mean efficiency.

Efficiency

The efficiency curve of a pelton wheel will remain slightly lower than that of a Francis turbine but the efficiency curve will be less affected by variations of the load. The balance in favour of the Francis turbine is further increased by the gain in the operating head, which amounts in fact to an augmentation of the overall efficiency of plant. Because of the wider latitude within which the velocity coefficient of a reaction runner can vary, the Francis turbine has a decided advantage as regards efficiency if the operating head varies appreciably.

Speed

For given operating conditions, the speed of a Pelton turbine could reach the lowest speed which can be considered for a Francis only by providing it with at least four jets. Where large outputs are considered, besides the inconveniences inherent to a four-jet turbine, it is common knowledge that high specific speeds are inappropriate for multiple-jet turbines, with the consequence that Pelton turbines will always run at an appreciably lower speed than a Francis designed for the same conditions.

Dimension

With the higher speeds obtainable with Francis turbines it follows that their overall dimensions will be appreciably smaller than those of the equivalent Pelton machine. Moreover, the water admission to the turbine (spiral casing and guide apparatus) surrounds the Francis runner much more closely than the supply pipes of the Pelton which are necessarily designed with

gently radiused sweeps to obtain the regular, easy curves to reduce as far as possible the rotational component of the velocity, which is very detrimental to the quality of the jet. For similar reasons the water velocity in the lead-in pipe of a Pelton turbine is generally kept much smaller than in the spiral casing of a Francis turbine and this naturally leads to larger diameters and consequently to a further increase of the dimensions of a Pelton turbine.

Wear

This problem concerns water-turbine practice as a whole but is of particular importance in the case of highhead machines, especially if the water contains abrasive sand.

Pelton turbines are sensitive to the wear of the needles, nozzles and of bucket splitters. It is easy to understand that with the high water velocities which occur in a Pelton bucket the least roughness of the nozzle and needle surfaces will cause a distortion of the jet which will cause it to spray, lose its compactness and to impinge on the buckets in conditions that are very detrimental to the efficiency of the machine.

As regards the runners, the Francis is favoured in that the water velocities are considerably lower and the bucket curvatures much less sharp. The weakest point of Pelton buckets, with respect to wear and its influence on efficiency, are the lips and the tips of the splitters. The pelton turbines require constant care and repairs or otherwise the turbine efficiency will fall away very rapidly.

Conditions are much more favourable in a Francis runner where the stream of water is perfectly guided and enters the blading at a much lower velocity; the runner blades are streamlined and have thick rounded inlet edges which permit considerable variations of the angle of approach of the stream of water without detriment to the efficiency.

Francis turbines are, however, sensitive to the abrasive effect of sand passing over the cheeks of the distributor, between these cheeks and the guide vanes and through the runner clearances; this wear can however be met satisfactorily by judicious design of the runner clearances and by the use of appropriate materials.

Maintenance

The overhauling of a Francis turbine generally entails the adjustment of the runner clearances by replacing the stationary rings and touching up the runner bands, or vice versa, and eventually the refacing of the distributor linings. It is very seldom necessary to repair the runner blading by welding and even then no annealing is required as local stresses have not the same importance as in Pelton runners.

The welding of the guide vanes requires no exceptional skill and can be done on site in the power house repair shop so that, really, the only delicate task required for the maintenance of a Francis turbine is the upkeep of the runner bands and stationary rings; no building up and reshaping of the hydraulic profile is required.

For a Pelton turbine, on the contrary, besides replacing the needle tip and the throat ring of the injector, each bucket must be required by welding over the whole or part of its surface and the splitter and cutout must be carefully built up and reshaped. The buckets must then be ground down to their original shape; this work can only be done by hand and requires careful checking by means of the original templates. After grinding it is advisable to anneal the runner so as to relieve local stresses caused by the welding and to make a careful inspection for shrinkage cracks etc. This work is of extreme delicacy and requires a great deal of skill; moreover it must be frequently repeated if the efficiency of the turbine is to be maintained within a reasonable measure of the initial figure. The relative simplicity of access and dismantling which is generally a feature of Pelton turbines is therefore more than counterbalanced by the frequency with which overhauls are necessary. On the contrary the efficiency of a Francis turbine is not nearly so sensitive to wear, moreover it is fairly easy to take steps to minimise the time required for replacing those parts which are subject to wear.

Regulation

All other factors remaining equal when full load is thrown off, the momentary speed rise will be about 50 per cent higher for a Francis unit than for a Pelton Unit.

To sum up, the points in favour of the Francis turbine are a more complete utilisation of the head and higher operating efficiencies for loads ranging between half and full load.

The gain in the utilisable head is undeniable, and is entirely independent of the load. The question of the efficiencies deserves to be examined in each particular case, according to the load factor of the machines. With plants having storage reservoirs working according to predetermined programme it is generally possible to operate Francis turbines at loads within the range of their maximum efficiencies. The Peltons have the advantage when one units of a power house has to serve as standby and to run at small load for long periods or when in run-of-the river plants, the available discharge remains for long periods appreciably lower than the discharge required to run the turbine at half load.

The speed of the Francis will always be higher than that of the Pelton with corresponding reductions in the cost of the generators and in the dimensions of the units, the machine hall and the power house equipment.

The efficiency of a Pelton is maintained in the neighbourhood of its original values at the cost of frequent overhauling and repairs.

The satisfactory operation of numerous high-head Francis runners has proved that this type of turbine is perfectly suited to heads ranging between 1000 meters and the experience gained with these machines leaves nodoubt that the limits they have now attained may be appreciably increased in the future.

A Comparative Study of the Kaplan and Deriaz Turbines

By P. DERIAZ

The Deriaz turbine is the outcome of research and development to provide a turbine, operating under heads in excess of 200 ft which would have the flat efficiency (load characteristic as well as the part gate stability of operation of the wellknown Kaplan turbine. It is well known that at heads above 200 ft the application of the Kaplan becomes extremely difficult both hydraulically and mechanically and so the problem is solved by means of a mixed flow variable pitch blade runner. The history of this work has been published[1]).

The first application of the new design was at the Sir Adam Beck-Niagara Pumping Generating Station of the Hydroelectric Power Commission of Ontario, where the units are required to run both as pumps and turbines. Six machines are installed with a total pumping capacity of 300,000 HP, See Fig. 1[2]. At present three Deriaz turbines of 110,000 HP each are being designed for a Spanish power station. These also will run in reverse as pumps. See Fig. 2.

The author finds, as a result of detailed study, that for heads in excess of 120 ft the mixed flow Deria turbine has marked advantages over the axial flow Kaplan turbine. With heads lower than 120 ft the higher speed of the Kaplan and consequent cheaper generator are predominating features in favour of the Kaplan.

To illustrate the advantages of the Deriaz machine, the remainder of the present paper sets out the results of the study made recently to compare a Deriaz turbine with a Kaplan for a specific installation. The design data is the same for both machines, as follows:—

Output 30,000 HP; Head 180 ft; Speed 300 rpm
Specific Speed n_s =350 (metric)

The cross-sectional arrangements of the two designs are shown in figures 3 and 4. It can be seen that they have the same guide vane pitch circle diameter, although the Kaplan has a some what larger gate height.

On this design the axis of the blades on the Deriaz machine are inclined at 30° to the horizontal which differ from the 45° hitherto adopted at Niagara. (Fig. 1). This is in order to suit a higher specific speed for this particular turbine.

We shall now consider the relative merits of the two alternative designs as shown in fig. 3 and on fig. 4 from the hydraulic and mechanical points of view.

HYDRAULIC CONSIDERATIONS

In comparing the hydraulic profiles of fig. 3 and 4 it will be noticed that fig.3 is better streamlined, i. e. offers a more direct passage from the gate apparatus to the draft tube.

Because the runner hub in fig.3 is much shorter than in fig.4 the wetted perimeter is reduced and a reduction in friction losses results.

The head of 180 ft which is high for the Kaplan calls for a relatively low specific speed. The radial extent of the blade $\triangle R_1$ should accordingly be small. This is readily achieved on the Deriaz without an increase in the water velocities because the mean radius at blade entrance is larger. In the case under consideration the radial extent $\triangle R_1$ of the blade on the Deriaz machine is 8% smaller than on the Kaplan. It should also be noted that while both designs have the same runner outlet area, the inlet area on the Deriaz is 27% larger than on the Kaplan. Low inlet water velocities are thus achieved.

Inlet and outlet velocity triangles for the mean blade section are shown for both Deriaz and Kaplan in figure 5. The increased inlet peripheral speed u_1 of the Deriaz combined with reduced inlet water velocity c_1 make the inlet and outlet relative velocity vectors w_1 and w_2 almost parallel, i.e. the inlet and outlet blade angles β_1 and β_2 are the same. For the Kaplan w_1 and w_2 are divergent, as the triangles show, β_2 β_1. When the blade is rotated, the inlet and outlet blade angles change simultaneously by the same amount, i.e. on the Deriaz they remain parallel, while on the Kaplan they remain divergent. As the flow conditions require that both β_1 and β_2 tend to zero when the flow reduces to zero, the Deriaz Runner remains in accordance with the hydraulic requirements at all blade angles. This results in a higher efficiency than the Kaplan at very reduced loads.

Compared with the Kaplan, the Deriaz machine has a reduced mean diameter at outlet for the same outlet area. This is the result of the smaller hub diameter at blade exit (D_{2i} see fig. 3 and fig. 4). This reduces the peripheral speed u_2 and hence the relative water velocity w_2. An improved cavitation factor results.

The cavitation factor is also improved by the angles β_1 and β_2 being equal. This means that the lowest absolute pressure zone is no longer on the convex side of the blade as in the Kaplan (necessitated by the deviation from β_1 to β_2) but is displaced downstream of the runner exit.

At abnormally high speeds the mixed flow design

tends to reduce the discharge. This effect is known on a Francis turbine of low specific speed, and gives a corresponding low runaway speed as compared with the Kaplan, where the axial flow results in high over-speeds and in high discharges at runaway. The reduction in runaway speed results in a cheaper generator for the mixed flow design.

Due to its larger hub diameter at blade inlet, the Deriaz runner produces a lower hydraulic thrust than the Kaplan. This is caused by the seal of large diameter (5) on the largest hub diameter (fig. 4) as compared with the small diameter of the shaft gland on the Kaplan (fig. 3). The space between seal (5) and shaft gland is connected to the tailwater on the Deriaz design.

With the relatively high head under consideration it is essential that the hub be of spherical form to prevent gaps between the blade and hub when the former is rotated. In the Kaplan version, this spherical hub produces a marked change in direction of the flow lines at the low pressure zone and thus there is a further risk of cavitation. The larger Deriaz hub leads to a larger spherical radius and a less pronounced change in direction, which is at the same time well upstream of the low pressure zone. The cavitation at the hub is therefore eliminated.

If operation as required then the mixed flow design is able to produce a very much higher head than the axial flow design. This is illustrated in fig.6. This figure was published in the paper by the author at the World Power Conference of 1958,[3]) but has since been revised in line with recent studies. These have shown the possibility of extending the application of mixed flow design to much higher heads than was then anticipated. The dotted zones in fig. 6 are an indication of the extension in head that are considered practical, both for turbine and pump operation.

MECHANICAL CONSIDERATIONS

The high head calls for a greater number of runner blades with relatively large trunnions to support the hydraulic forces. In the present study 10 blades are accommodated in the Deriaz version against 8 blades for the Kaplan. This is achievable without difficulty in the larger hub.

The inclined position of the blade trunnions results in the centrifugal force F from the blade (see fig. 3) having a momentum $F \times e$ opposed to the momentum of the resultant hydraulic pressure $P \times i$. It follows that the larger and more heavily loaded trunnion bearing is partly relieved. In the present study the relief amounts to 43% on the larger bearing and 13% on the smaller bearing.

There is no corresponding relief of bearing loading in the Kaplan version of fig. 4, because the centrifugal force is in line with the blade trunnion.

With the high heads under consideration, the operating gear common to the Kaplan turbines and illustrated on fig. 4 becomes very difficult to locate.

(*a*) There is very limited space between the large trunnion bearings and the operating levers (7) which are placed between these bearings, become extremely short, unless the hub diameter is increased much more than is desirable from an hydraulic point of view.

(*b*) The shortness of lever (7) gives rise to high forces on the links (8) and reduced strokes (9) of the operating rod (6) and servomotor piston.

With the Deriaz design illustrated in fig. 3 these difficulties are overcome as follows :—

(*a*) The levers (7) are at mid stroke position in the radial plane of the blade trunnions. They can thus be accommodated without difficulty. Their length can therefore be larger even with the increased blade number.

(*b*) There are no links like (8) of fig. 4. The transmission is direct through the sliding block (8) of fig. 3.

Fig. 7 describes a design sometimes used with Kaplan turbines to overcome the limitation on the dimension of the lever. Here the lever (2) is larger than the spacing of the trunnion centreline (1) would allow. This necessitates an intermediary bell crank lever (4) with fixed fulcrum (5) and two double-links (3) and (6).

All this gear has to be accommodated in the restricted space within the hub. The larger the number of clearanced pins and bearings, the less accurate becomes the control of the position of the runner blades. Considering the high degree of finish required and given to Kaplan blades, it does not seem satisfactory to introduce devices which allow for possible diversity in the angular position of the blades.

The servomotor of the runner blades is the item that differs mostly between the Deriaz and Kaplan designs. See fig. 3 and 4. With the Kaplan, the servomotor has a stroke (9) axial to the turbine shaft. The servomotor acan generally be placed as shown by (10) on the shaft where no diameter restrictions occur.

On many Kaplan designs this servomotor can be located in the position indicated by (11), i.e. inside the hub, thus aboiding the long and heavy piston rod (6). For high heads this is not possible since the diameter (13) exceeds the available space (11). One solution consists in raising the governor oil pressure to permit reduction in servomotor volume displacement, going to 400 psi which is not usual. Another system consists in building a double tandem servomotor in the space below the diameter (11) in the hub, which is very complicated.

On fig. 3 the Deriaz version shows the servomotor (10) of the moving vane type. The section AA is

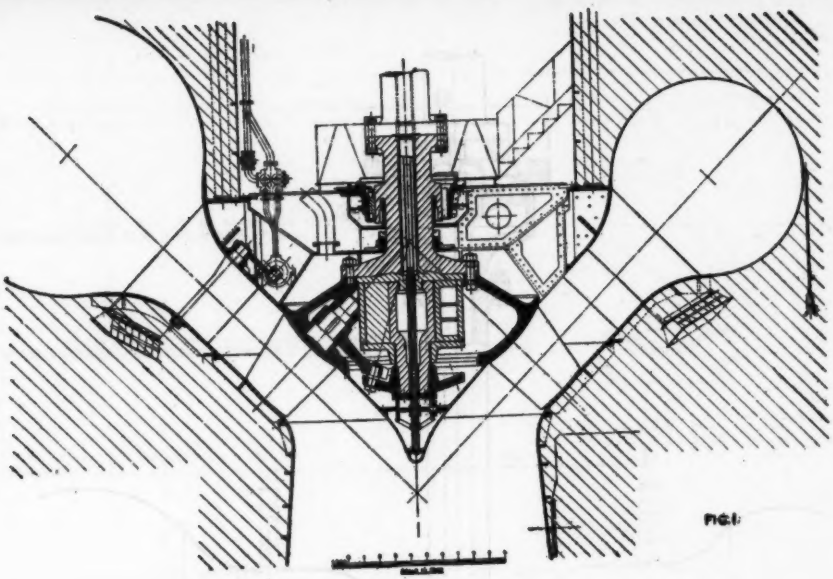


FIG.1

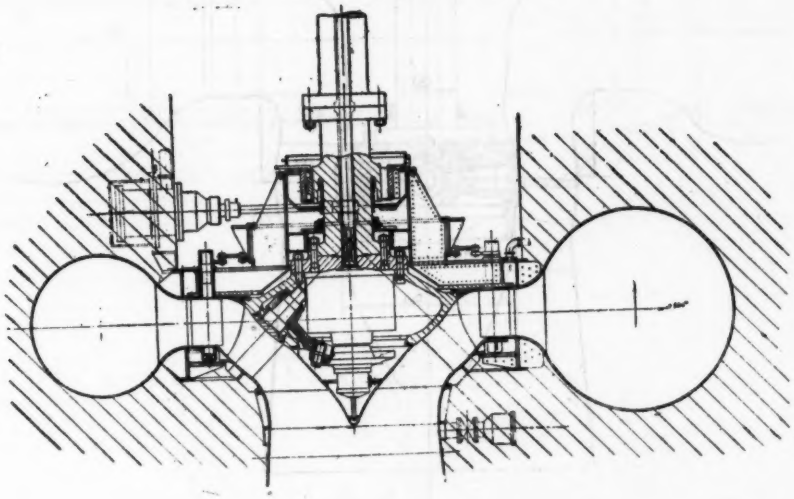
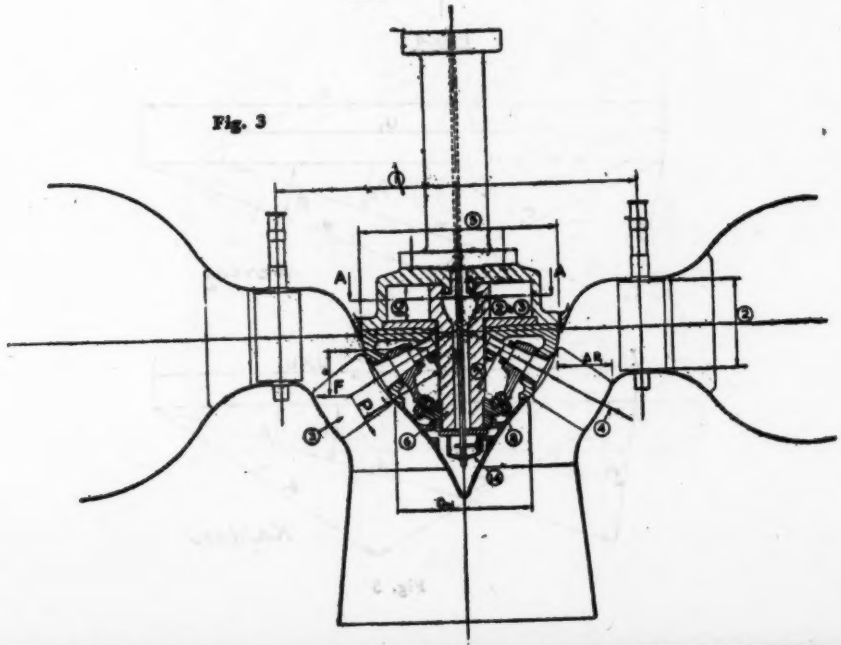


FIG.2

Fig. 3



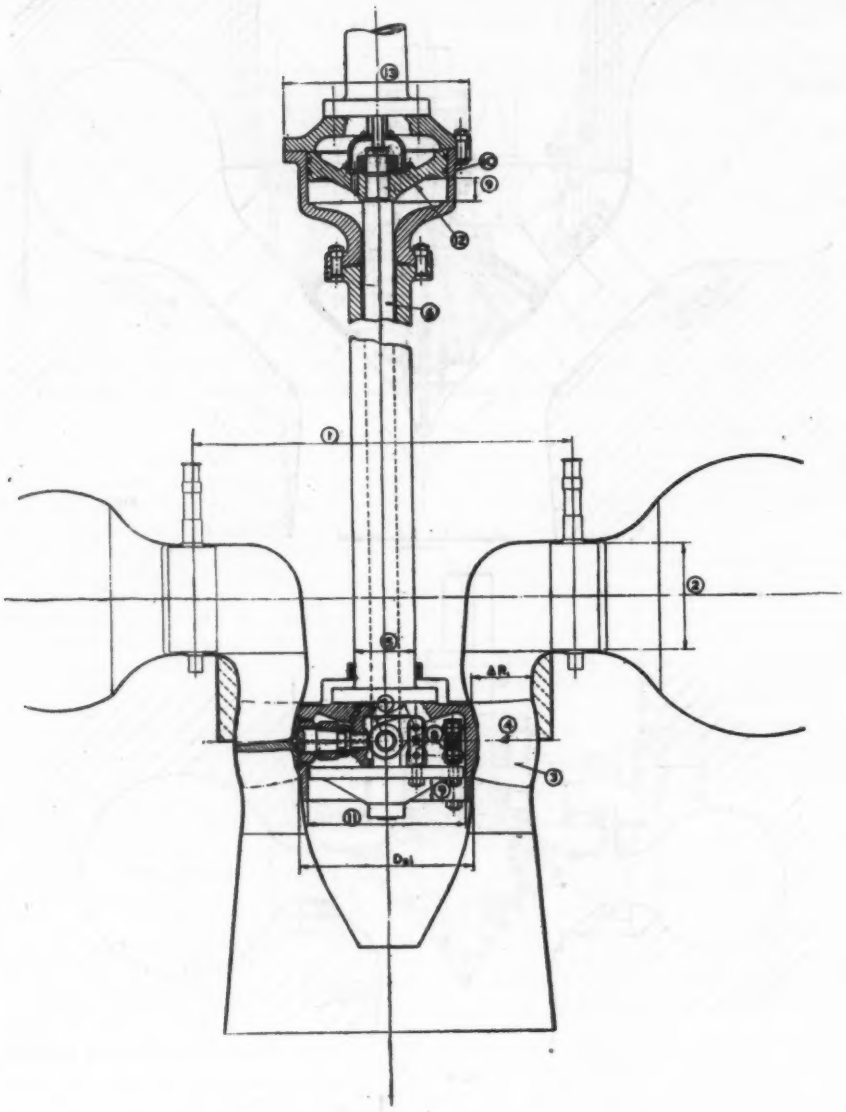


FIG. 4.

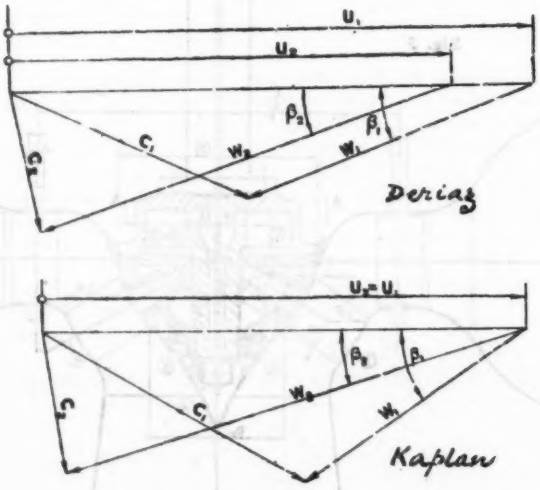


Fig. 5

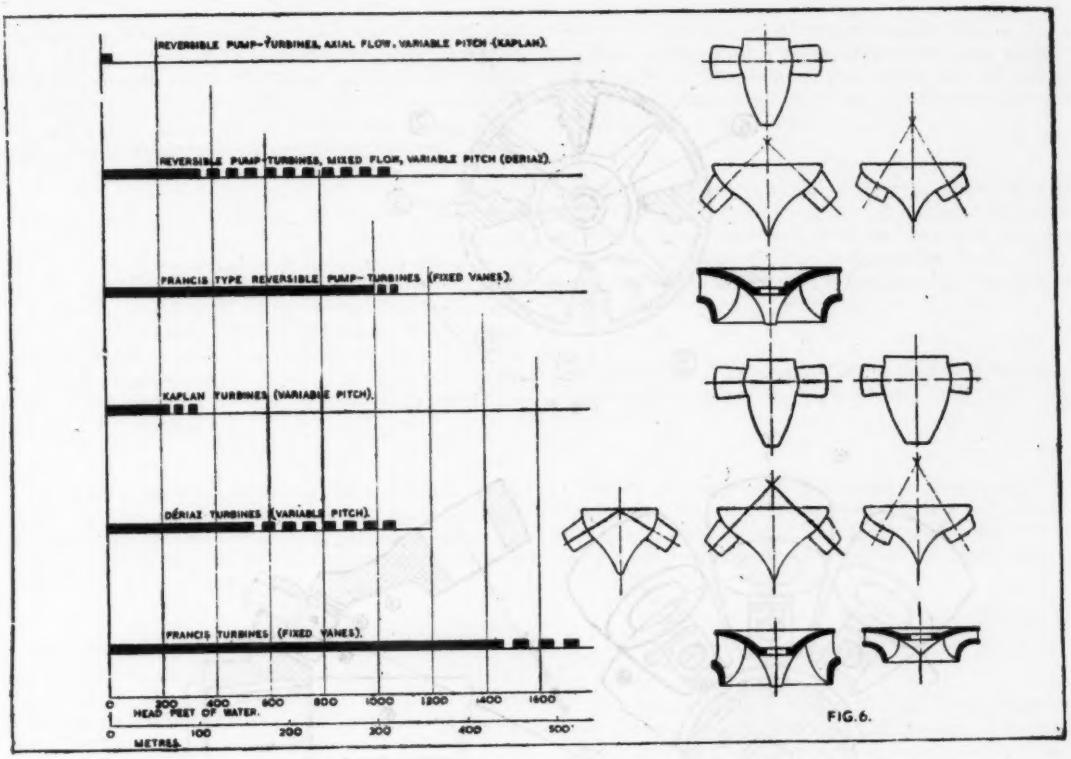

REVERSIBLE PUMP-TURBINES, AXIAL FLOW, VARIABLE PITCH (KAPLAN).
REVERSIBLE PUMP-TURBINES, MIXED FLOW, VARIABLE PITCH (DERIAZ).
FRANCIS TYPE REVERSIBLE PUMP-TURBINES (FIXED VANES).
KAPLAN TURBINES (VARIABLE PITCH).
DÉRIAZ TURBINES (VARIABLE PITCH).
FRANCIS TURBINES (FIXED VANES).
0 200 400 600 800 1000 1200 1400 1600
HEAD FEET OF WATER.
0 100 200 300 400 500
METRES.

FIG.6.

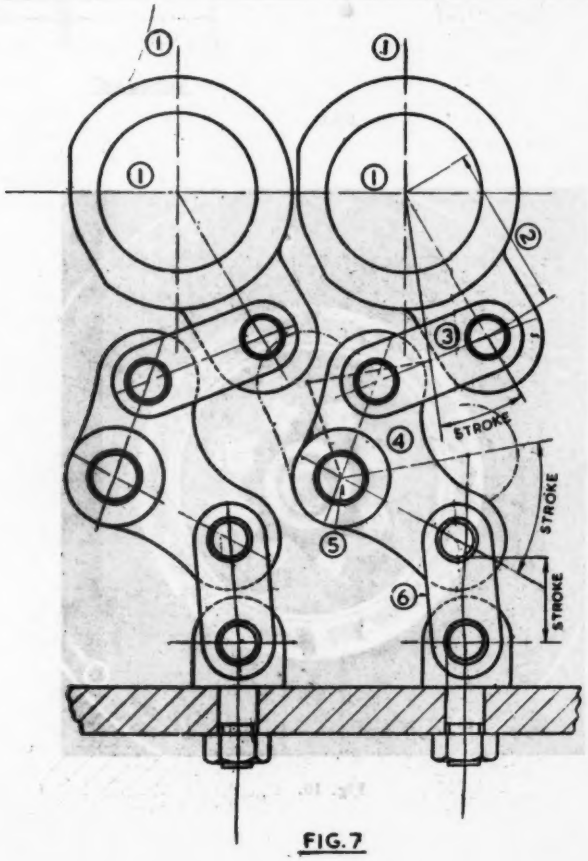

1
1
1
1
2
3
4
5
6
STROKE
STROKE
STROKE

FIG. 7

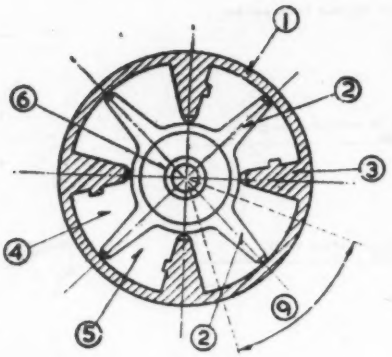


Fig. 8.

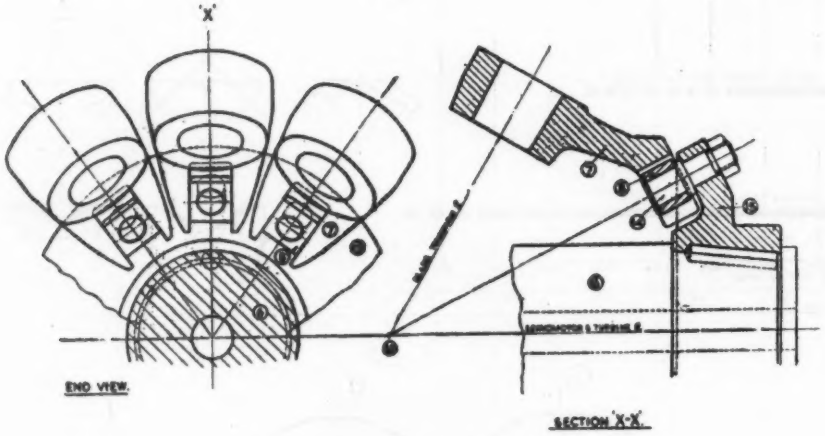


FIG. 9.

Fig. 10.

given on fig. 8 and shows four fixed vanes (3) attached to the body (1) and four movalbe vanes (2) attached to the operating shaft (6). These vanes separate eight chambers, four of which (left hand) give movement in one direction when filled, whilst the four others (right hand) are connected to exhaust, and vice versa.

The servomotor height (10) (see fig. 3) can be varied to suit the energy required to operate the runner blades without an increase in the overall diameter of the servomotor body. The shaft (6) operates over the angular stroke (9) of fig. 8. Its displacement is trasnsmitted to the levers (7) of fig. 3 by a mechanism which is given in further details on fig. 9. For clarity, the blade trunnion is not shown but only its centreline. Again on the end view the pins (14) have been removed for clarity.

The gear is drawn for position of mid stroke as for fig. 8. As the servomotor shaft (6) assumes other positions, the levers (7) incline as can be seen in the photograph of fig. 10 which shows the levers in "blades shut" position.

The geometrical requirements for correct functioning are that the centreline of the pin (14) and its sliding block (8) meet point (16) where the runner blade trunnion centreline and the vane servomotor centreline meet.

This construction is new and protected by patent.

The moving vane servomotor as per fig. 8 is somewhat more intricate to manufacture than a plain cylindrical servomotor. It brings, however, considerable simplification elsewhere and is thereby fully justified economically. It has proved entirely reliable in service.

When the Kaplan turbine is designed for high heads it is essential that the runner envelope be spherical both above and below the blade centreline in order that there should be no gap between blade and envelope. It follows that to dismantle the machine the upper half of the envelope must be removable. The shaded part of fig. 4 indicates the portion of the envelope that must be made removable. The Deriaz type runner is free from this difficulty. The spherical surface is built all along the runner skirt and at the same time this part can be connected in once and for all as it requires no subsequent dismantling.

Especially for high heads, gap cavitation requires close supervision. This cavitation is caused by the clearance between tip of runner blades and the fixed envelope surrounding them. With the axial runner (Kaplan) this clearance is fixed at the time of manufacture and cannot be controlled thereafter without modifications to the blade tips.

With the mixed flow design (Deriaz) this clearance is readily adjusted by setting the position of the runner in such a way that only just the required running clearance is allowed. This adjustment does not require any dismantling and is achieved by the setting of the thrust bearing pads.

It therefore follows that the mixed flow design is much better suited to high heads than the axial flow design.

Station Arrangement: The shorter runner hub and the lighter generator design permit the installation of a lower crane of smaller capacity with the Deriaz design. This leads to very valuable economy in building costs.

CONCLUSION

The present paper has outlined the advantages that result from the mixed flow design of variable pitch runners. It is the author's opinion that more and more applications will be found and realised in future. The field for these applications is not yet fully explored. Particularly in the reversible pump turbine field, as is required for storage of energy by hydraulic means, the mixed flow Deriaz design offers the only solution for a centrifugal pump that can be controlled in its power input over wide ranges and with high efficiencies. This is essential for high heads and very large size units. This consideration permits a high flexibility of service on the pump cycle comparable to that on the turbine cycle. It is this flexibility that has caused the hydraulic turbine prime mover to be so very greatly appreciated for peak load generation. It can now be obtained on the pump cycle.

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The Hydraulic Turbine in Evolution

By P. DANEL

In this paper an endeavour is made to give an appraisal of some aspects of the evolution of hydraulic turbines during the last 50 years.

It has been sometimes said that hydraulic turbines reached such a degree of perfection some decades ago that only minor changes can be expected in the way of improvements; contrary to that belief this paper will show that in certain respects spectacular changes have occurred in recent years with probably more to come in the near future.

It will be assumed here that the reader has at least an elementary knowledge of what are the various types of conventional hydraulic turbine; the impulse turbine or Pelton turbine; the Francis turbine; the propeller turbine (fixed blades); and the adjustable-blade propeller turbine or Kaplan turbine.

To these in recent years various types have been added using a runner similar to that of the Kaplan but put in quite different settings. Of these new types the most important is probably the Bulb turbine (also called Turbe turbine) which is briefly described below.

Several types have been designed so as to be used either as a pump or a turbine.

With the rapid development of electric power at the turn of the century, hydraulic turbines were designed for greater output and higher speeds.

However, for some years the maximum size of unit was limited by the possibilities of the thrust bearings then available, and most of these bearings were short-lived even on units of only 5000 or 10000 kW, which were then considered to be large ones.

The introduction of the oil-wedge type of thrust bearing using Osborne Reynolds' principle soon removed that important limitation.

Since the advent of electric power most laboratories devoted to hydraulic turbine research tried to develop faster and faster turbines so as to reduce the physical size of units and their cost. In recent years this has been the object of a spectacular advance especially in the field of low head developments and tidal power.

Before discussing the limitation encountered in developing high-speed units, let us survey briefly the results obtained in the last half century.

SURVEY OF RESULTS IN THE LAST 50 YEARS

In many previous discussions of this aspect the various hydraulic turbines are classified according to their specific speed although it is not usually made clear why such a parameter is adequate to characterize the various types.

The problem may be briefly stated in this way:

The specific speed is

$$n_s = \frac{n\sqrt{N}}{H^{5/4}} \qquad (1)$$

where n_s is specific speed, n rev/min, N power in h.p. (metric), and H head (metres).

The specific speed being a dimensional quantity, the system of units has to be specified. Here we will use the metric system and give the ft/lb equivalent

$$n_s \text{ (meteric)} = 4.46\ n_s \text{ (ft. lb)}$$

As given by equation (1) we see that the specific speed depends on three parameters so that the same value of ns can apparently be obtained in many ways and conversely the knowledge of n_s could not give an idea of the type of turbine.

However, geometrically similar turbines operating under similar conditions have the same specific speed. It will be sufficient then for the purpose of our discussion to consider turbines with a runner of unit diameter (1 m) operating under unit head (1 m).

The expression of the specific speed becomes

$$n_s = n_{1\text{-}1}\ \sqrt{N_{1\text{-}1}} \qquad (2)$$

where $n_{1\text{-}1}$ is rev/min of unit runner under unit head and $N_{1\text{-}1}$ is power in h.p. of unit runner under unit head.

The expression of the specific speed as given by equation (2) cannot be further simplified and we see that it is given by the product of two apparently independent terms.

If these two parameters were wholly independent, the same value of n_s could result from an infinite number of combinations of the two parameters. However, although they are not linked by a straightfoward mathematical function, they are none the less tied together by a number of conditions so that when the specific speed of a unit is known, the essential features of its hydraulic design are also closely determined.

Practically we can consider that $n_{1\text{-}1}$ and $N_{1\text{-}1}$ are related by an empirical function F. However, this function F proves to be different according to the type of turinbe so that the specific speed is a good parameter to compare two turbines of the same group but may lead to false impressions when used to compare turbines of totally different types. Fig. 1 has been prepared to show the present trend of this empirical function F for both Francis and Kaplan units.

The unit speed $n_{1\text{-}1}$ is chosen as the abscissa and the unit power $N_{1\text{-}1}$ as the ordinate.

Curves of equal specific speed are shown with the value of ns in the metric system on the top left and with the equivalent value in ft lb at the bottom right.

A strip marked Francis and limited by two lines shows the present trend of F for Francis turbines; the lower line corresponds approximately to the maximum efficiency and the top line to the rated power. With modern units there is little scatter and the general trend would be similar whoever the manufacturer may be, although the resulting curves might not be indentical. We have already pointed out that F is not a straight forward mathematical function. It is actually a compromise between many practical conditions. Before we discuss some of the more important of these, let us remark that most research is concerned with large units and some conditions which may be imperative for a large unit will bear largely on all this research and indirectly influence the design of smaller units for which it is usually uneconomic to do any costly research.

Let us now review some of the most important conditions that have a bearing in determining the narrow band in Fig. 1 where most modern Francis turbines plot.

One of the most obvious conditions is that the turbine must be efficient, with a high peak efficiency and a sustained efficiency at both part load and overload as far as possible. This single aspect of the problem will restrict the possibilities to a band having the general characteristics shown in Fig. 1, but it would be much wider than that shown. With sufficient research and testing it is possible to depart widely from the curve and still show a high efficiency as the design of special purpose units has often shown. Practically, the efficiency condition will not be the ruling factor in determining the F function.

A more important condition is economic, especially for large units it is essential that the overall dimensions and total weights of units be as small as is consistent with a good performance. Good performance means not only good efficiency but also ease of maintenance, no vibrations, no cavitation.

It will be found that little or no cavitation is the most important condition. Let us compare for instance three Francis units of the same specific speed (say, 70 metric), well designed as far as efficiency is concerned and whose peak efficiency would be respectively located at A, B, and C (Fig. 1). The unit at B has a much higher unit power than that at A, which also means a higher unit discharge. This in turn means higher absolute velocities in all the water passages with increased cavitation danger, especially at the runner outlet and for high loads.

On the other hand, the unit at C has a much lower unit power and discharge than that at A but a much higher unit speed; this in turn means higher relative velocities in the runner creating a cavitation danger in the runner, especially at part loads. (A complete discussion of this problem would carry us too far.)

It thus appears that if we depart from the F function the cavitation danger increases both ways, with higher $N_{1\text{-}1}$, the cavitation danger increases rapidly especially at high loads and with lower $N_{1\text{-}1}$ at part loads. A more complete discussion would show that to have a minimum cavitation danger only a narrow strip of Fig. 1 is available so that the cavitation condition practically determines the whole trend of the function F. Whatever margin may be available is made use of to satisfy economic conditions.

The natural evolution in the last 50 years has been toward a gradual narrowing of the available strip so that Francis turbines of the same specific speed have a tendency to have almost geometrically similar runners, not because manufacturers copy each other but because of complex conditions which are gradually becoming better known.

Our discussion so far has dealt primarily with Francis units; a similar condition obtains for fixed-blade propeller, Kaplan and Bulb units—giving a similar trend, the F function crossing the graph with a positive slope. In Fig. 1 the F trend for Kaplans has been shown. To avoid crowding the graph the fixed propeller trend has been omitted especially as very few such units are now being built in France.

It will be noticed in Fig.1 that if the F functions for Francis and Kaplan units show a similar trend they are not identical. With present types of unit there is quite a jump from a Francis of $n_s = 350$ to a Kaplan of the same specific speed, the Kaplan having a much higher unit speed. The old Lawazeck type units would be in between. These had been abandoned for a long time until recently, when they have proved interesting for mediumhead units having to run part time as a turbine, part time as a pump.

The F function for impulse or Pelton wheels has not been shown in Fig.1 because the scale chosen proves inadequate. Practically, for large modern Pelton wheels, the unit speed varies only from 38 to 43 while the unit power varies from 0.6 to 0.05 so that the strip representing Pelton wheels in Fig.1 would be almost vertical but with a negative slope.

Let us look in a more detailed way into some of the conditions which contribute to determining the F function for Pelton wheels.

To each design of bucket corresponds a family of different Pelton wheels obtained by varying the pitch diameter and the number of buckets.

In Fig. 2 the peak efficiency and rated power of two such families are shown. To avoid overcrowding the figure, only two representative families are shown and for clarity a very large scale has been taken for the unit speed. A different bucket design will be chosen out, however, for a medium-head or a high-head Pelton. Some of the conditions to be considered will now be briefly mentioned.

Cavitation both of the bucket splitter and of the back should be avoided; this will limit the specific speed per jet according to the head, the present trend being shown in Fig. 3.

Fatigue of the bucket metal will introduce another important limitation of the possible maximum number of impacts per minute per bucket for a given head. Fig.4 gives an example of such a condition for two types of buckets.

We now see that for a large Pelton wheel of a given specific speed, the choice of bucket design and the number of buckets will be rather restricted so as to fulfil many conditions, so that the specific speed almost describes the hydraulicdesign of the impeller.

We thus have seen that the specific speed, although depending on two parameters is practically sufficient to describe the hydraulic design of current types of turbine, Pelton, Francis, Kaplan, etc.

Let us now see how the peak efficiencies of the various types of turbine vary according to their specific speeds.

Such a comparison was made years ago by Graf and Thoma and is reproduced in Fig. 5.

To determine their curve, the authors selected a number of European turbines. Their curve was intended to give the maximum efficiencies obtainable. The theory of hydraulic turbines as it then existed enabled the design engineer to estimate the various losses through the various water passages of the machine. However, this theory was semi-empirical, and many coefficients had to be adjusted by comparison with a few existing turbines. When such an empirical approach is used, results cannot be extrapolated with any chance of accuracy, and even when interpolating some of the defects of the original turbines from which the coefficients were adjusted are likely to be embodied in the design. In extrapolating, such defects may be exaggerated. No wonder then that Graf and Thoma were wrong in assigning a ceiling to the performance of hydraulic turbines.

It should be noted here already that at the time of publication of Graf and Thoma's paper there were already quite a few hydraulic turbines both in Europe and in the United States with efficiencies above that of their curve. Either these units were not known to these writers or they did not believe the results for some reason or other.

In delving through old archives I have found a few instances of high performances of then unconventional units, performaces which were not believed in at the time and the design not followed up until rediscovered many years later. Even in research it may prove difficult at times not to be the victim of traditional ideas.

Professor Ernst Reichel published a new paper on the same subject. He gave two curves, which are reproduced in Fig. 5. Curve A represents the progress made in Germany since the Graf and Thoma paper was written. The second curve marked B corresponds to the performance of a model unit designed by Wagenbach and gives the highest specific speed attained in Germany at the time.

In August 1909 Chester W. Larner published a paper on the characteristics of modern hydraulic turbines.

In a written discussion contributed to that interesting paper Lewis F. Moody and John C. Parker, after recalling both Graf and Thoma's curve and Reichel's, gave their own curve based chiefly on many tests of model runners at the then famous Holyoke laboratory as well as on a few units in the field. This Moody and Parker curve already lay well above the previous curves, although it drops rapidly toward the higher specific speeds.

In the few following years in both Europe and America many turbines were designed with peak efficiencies above Moody and Parker's curve. It should be noted that Moody himself designed several of these with performances above the limits he had set himself.

In 1921 Moody published a new paper giving a new limiting curve (Fig. 5) showing the tremendous progress accomplished in those 12 years.

This curve embodied results from various manufacturers and many of the units were larger than those considered previously. This explains in part the higher results due to the now well-known step-up in efficiency with size. This 1921 curve was intended as a ceiling to possible performances.

Discussing the same topic in 1929, Professor Moody gave a new curve to take account of the progress made in the past eight years by various designers, including himself. He there gave a curve which he thought was not likely to be surpassed for many years to come (Fig. 5). In a personal discussion with the present writer, in 1936, Professor Moody still expresed the same opinion and, with a certain sense of humour, alluded to his previous curves and the cause of the error of assigning a ceiling to performances when dealing only with semi-empirical theories and expreimental statistical data.

Actually for more than a decade few field tests surpassed Moody's curve of 1929 and for some time it could be believed that it was a real physical limit to hydraulic turbine performance. Some manufacturers even almost abandoned research, feeling that little remained to be gained. This, however, was not the universal opinion and some laboratories have been very active in the last 15 years on various research projects pertaining to hydraulic turbines.

PRESENT TRENDS

The progress in recent years has been quite noteworthy on many design aspects. The present discussion is limited to studying only the maximum efficiency variation against specific speed, a curve is given in Fig.5 showing the present trend for Francis, Kaplan, and Bulb turbines. The small circles represent a few selected field tests. Some are already above the given curve which is not here to be taken as a ceiling to maximum performance; rather it is intended to show the present trend with an even chance for a good design to be above or below the curve in question. For the sake of comparison all results have been recomputed by the Moody step-up formula so

that the Francis curve corresponds uniformly to a 4-m runner and the Kaplan and Bulb curve to a 7-m runner, these figures being taken as average sizes for modern large units. It could compare with Moody's curve for 1929 which already embodied field tests of many large units. However, Moody's 1929 curve was still intended to show a maximum or ceiling which is not the case with the curve given for 1958: the latter merely indicates present trends.

If we now look at Fig.5 it appears that the most spectacular progress has been in the perfecting of the Kaplan turbine in the 1920's and that of the Bulb turbine in recent years. Progress in Pelton and Francis turbines has not proceeded by big jumps but was none the less continuous, so that before discussing recent advances in high specific speed units a brief review of progress in other types will first be given.

The Graf and Thomapaper had already given a curve (Fig.5) for impulse wheels with a peak efficiency at 86 (the original Pelton efficiency in 1888 was in fact 82).

At the present time, large Pelton wheels frequently have efficiences between 90 and 92, showing the considerable progress accomplished in 50 years.

Some 40 years ago, small Pelton wheels were developed with specific speeds as high as 36 metric; these were not very efficient and have now been abandoned for multiple jet Pelton or for Francis turbines.

Modern large Pelton wheels seldom now have a specific speed per jet of more than 30 and that for relatively low heads (less than 350 m) where competition from Francis turbines is severe. The field of large Peltons is covered generally by specific speeds ranging from 10 to 28 (metric). For example, large units under 1000 m may have a specific speed of 18-19 (metric).

The present trend is characterized by a gradual replacement of separate bucket impellers by integrally cast ones. This has been made possible by the steady progress in steel foundry work in the past 30 years and has enabled the manufacture of certain types of turbine which would have failed through metal fatigue with the separate bucket design. Multiple-jet units have been designed for many years, most of the early types being horizontal shaft, 2-jet units. In recent years many large vertical-shaft, multiple-jet units have been built. The number of jets is now of ten as large as 4, 5, or 6, with the same peak efficiencies as the best single-jet units. With multiple-jet units the specific speed is multiplied by the square root of the number of jets, for example, some recent 6-jet units have a specific speed as high as 55 (metric).

Some early Francis designs had a specific speed under 50 but their peak efficiency was lower than 83; with the progress of faster wheels, these could no longer be considered.

Modern large Francis turbines show exceptionally high efficiencies, often above 94 from a specific speed of 80 to approximately 350 (metric). Below 70 little progress has been made in recent years, not necessarity because of some fundamental physical reason but because research has not been very active in this field.

Specific speeds from 10 to 55 (metric) are well covered by good Pelton wheels and good Francis turbines start, say, at 75 (metric). It thus appears that specific speeds from approximately 55 to 75 are not covered by either Pelton or Francis. This has been termed by some authors the gap between Pelton and Francis. In real practice there is no gap and this very notion of a gap is and has been for the last 50 years a false problem.

The basic data for a hydro-electric project are the available head and available flow. The number of units and their speed has to be chosen knowing the conditions and limitations of each type of turbine.

As we have seen the F function for Francis turbines has no relation with that of Pelton turbines, so that when the relative merits of either Pelton or Francis have to be compared for a given project, different speeds will be chosen. In current practice, there is quite an overlap between the field of Pelton and that of Francis turbines.

This overlap was discussed at some length by A. Puyo in 1949. He showed how complex the problem of choosing between either type may be in certain cases. He showed also how this choice affects the number of units in a given plant.

He thus determined approximately the field of possible overlap between the two types (Fig. 6). Since his paper was published both Pelton and Francis turbines have made steady progress and Fig. 7 is offered as giving the present trend.

It must be remarked that the overlap is now much larger than ever before. Most of the arguments in favour of either as given by Puyo would still be valid only the numerical boundaries have changed. Those given in Fig.7 show only a trend and a few stray units could be already cited, the tendency being to widen still more the field of overlap.

To summarize briefly, the Pelton is usually sturdier, easier to operate and maintain, but costs a little more. The cheaper Francis is not so easy to operate and maintain and has more difficult problems in governing, leading to larger values of the water hammer during transients, so that the penstock may cost more.

One of the important factors will be the silt content of the water. With almost pure water without silt or grit the advantages of the Francis will appear more tempting, but as soon as the water contains any amount of abrasive which cannot economically be removed, the Pelton may be preferable.

Before the advent of Kaplan turbines Francis turbines had been developed for specific speeds as high as 450. Now with the perfecting of the Kaplan, Francis units are no longer built with specific speeds of more than 350. This value of 350 is not, however, a fixed limit between Francis and Kaplan turbines; here again there is quite an overlap.

In recent years Kaplan units have been built for higher and higher heads with corresponding lower specific speeds. Kaplan units have been built currently in recent years for heads between 30 and 50 m, which

The Hydraulic Turbine in Evolution

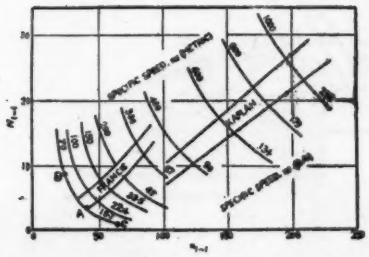


Fig. 1. Present Trend of the Empirical Function F for Both Francis and Kaplan Units

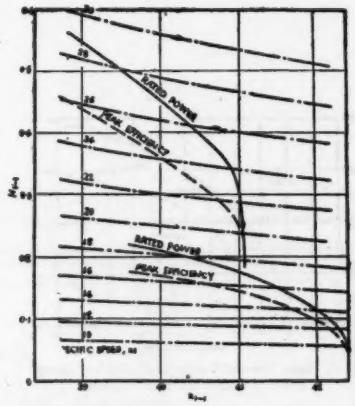


Fig. 2. Peak Efficiency and Rated Power of Two Families of Pelton Wheels

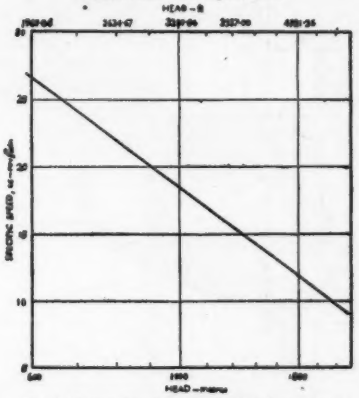


Fig. 3. Present Trend of Specific Speed and Head Relationship

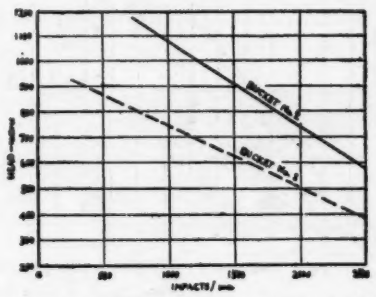


Fig. 4. Possible Number of Impacts per Minute per Bucket for a Given Head

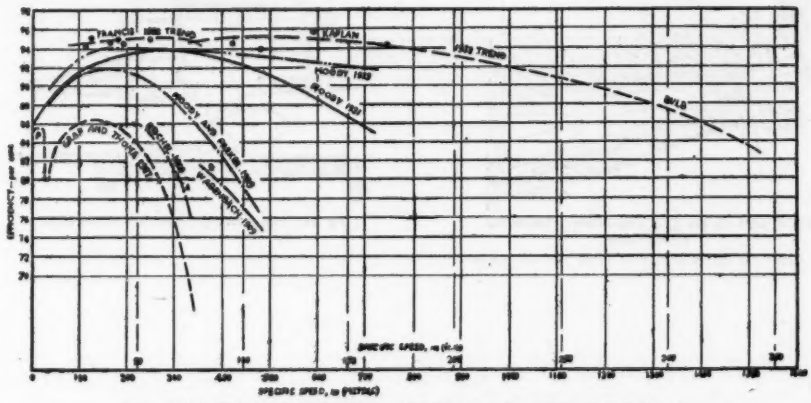


Fig. 5. Peak Efficiencies of Various Types of Turbine According to Their Specific Speeds

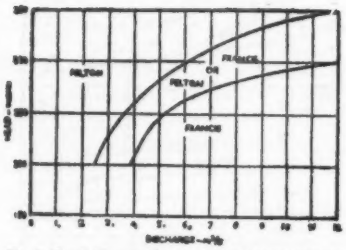


Fig. 6. Graph Showing Overlap in the Fields of Specific Speeds of Francis and Pelton Turbines

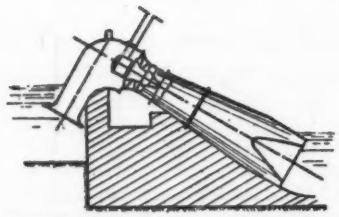


Fig. 9. Small Siphon of La Magnanerie on the Mayenne River

N 2.25 m. *Q* 4000 l/s. *N* 89 h.p. *n* 224 rev/min.

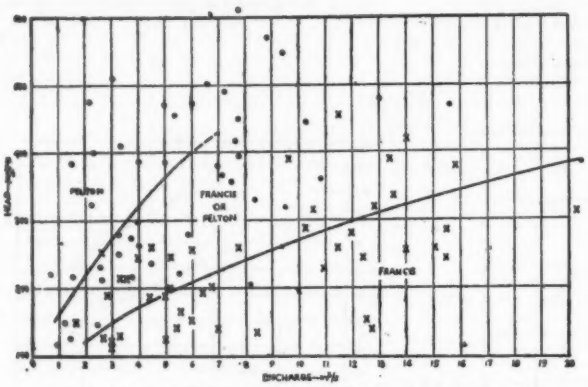


Fig 7 Comparison of Present Trend in Overlap for Francis and Pelton Turbines

o Pelton. + Francis.

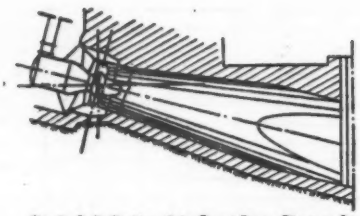


Fig. 8. Bulb Turbine of the Coastal Pump Plant on the Gave d'Oloron

H 7.6 m. *Q* 12.500 l/s. *N* 2100 h.p. *Tr* 250 rev/min.

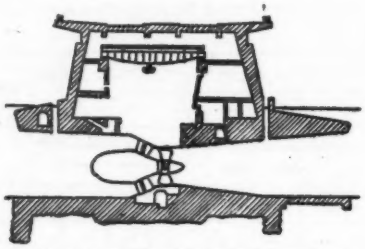


Fig. 11. Cross-section of Future Tidal Power Plant

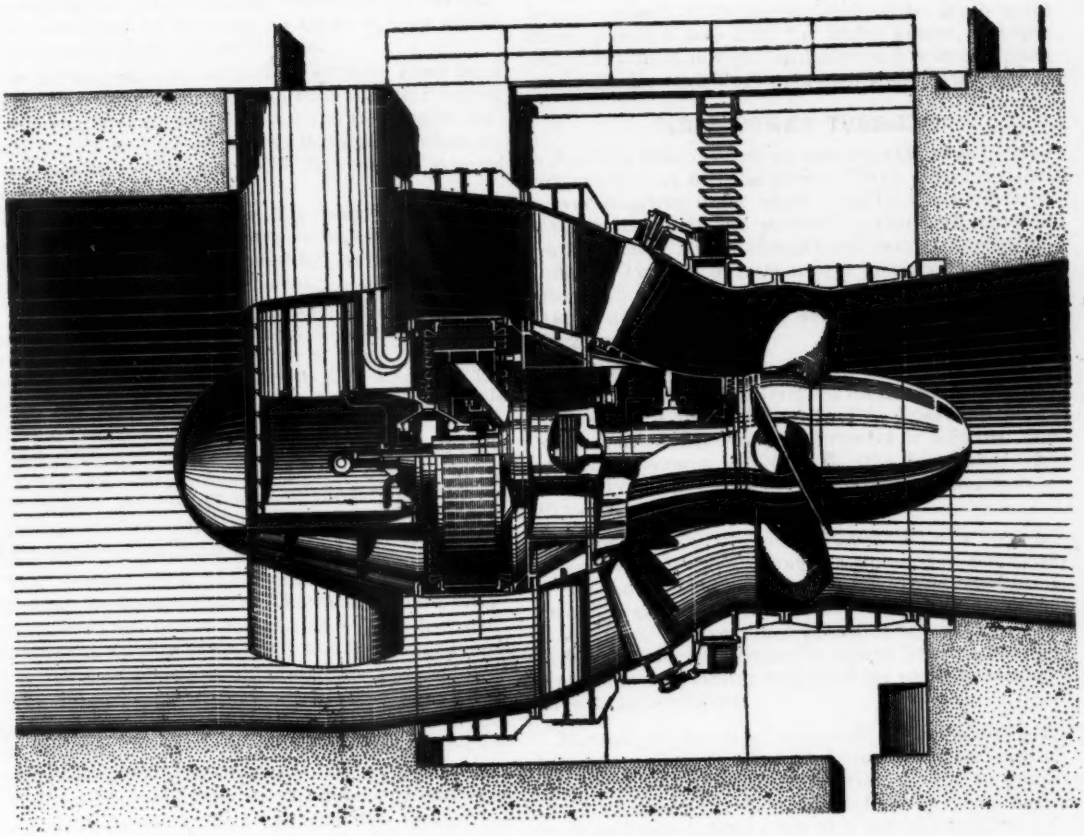
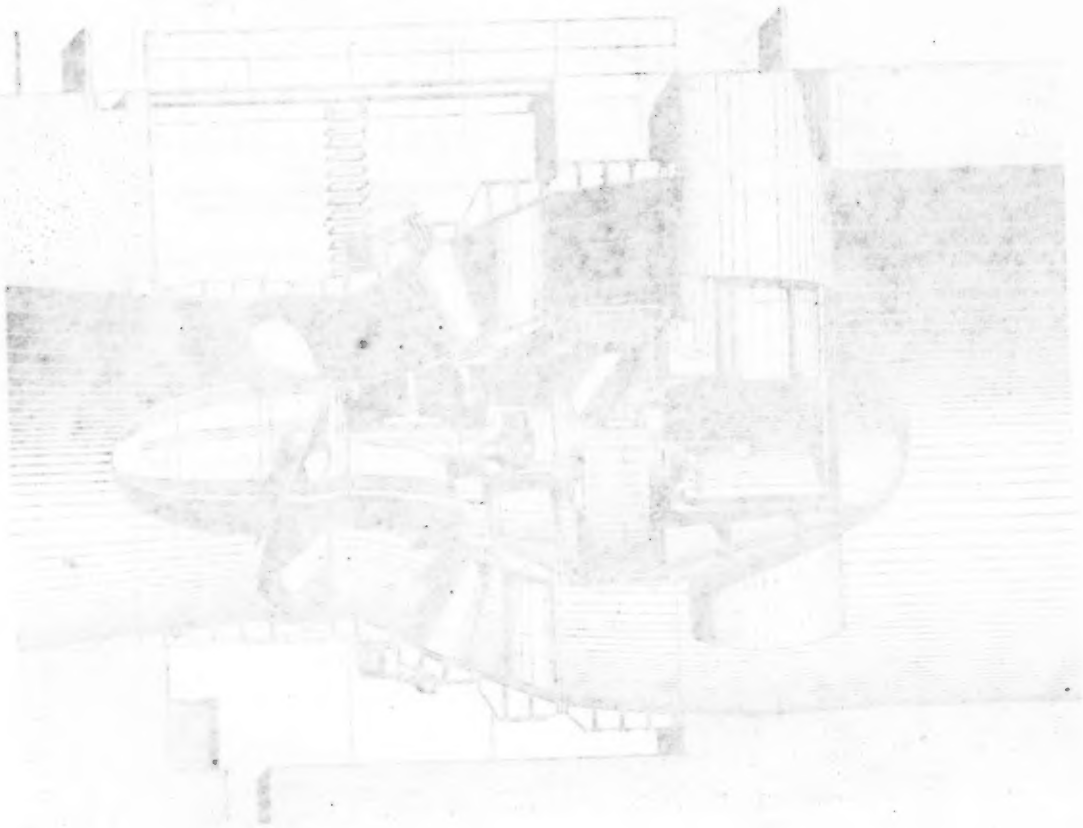


Fig. 10. Tidal Power Turbine



would have been considered very difficult in the 1920's or 1930's. Under exceptionally good conditions, a Kaplan unit has been built at Bort-Rhue operating under a maximum head of 80 m (244 ft) the specific speed being then as low as 227 which is very low indeed for a Kaplan.

The Moody curves for 1921 and 1929 (Fig. 5) were limited to specific speeds of approximately 700 (metric). Through the years there has been little progress; a Kaplan would be recommended now a days only for specific speeds upto 700 or 800 (metric). Although some Kaplans could eventually run at slightly higher specific speeds, especially at overload, they would not be recommended now in view of the advent and progress of Bulb units.

As already mentioned various settings have been successfully used with a Kaplan-type runner; it would be too long to discuss them here and the present discussion will be limited to the Bulb unit, reference being made to Gariel's paper for a more general discussion of various types.

The Bulb turbine was initially designed as a small unit to be set within a pressure pipe with a small generator in a bulb surrounded by the flowing water. These small units were inefficient.

The modern Bulb unit is no longer in a long pipe, and the water passages are kept as short as possible to reduce friction losses. After a short converging section at the inlet comes the bulb containing the generator, then the adjustable-blade runner and lastly a well designed draught tube.

The generator rotor may run in oil for small units in compressed air in larger units, or even in air at atmospheric pressure in some of the largest units.

In Fig. 8 and 9 are represented two early Bulb units of the modern family of efficient turbines.

Fig. 8 shows the Bulb turbine of the Caster Power Plant on the Gave d'Ossau. The turbine and generator were shipped to the plant as an integral unit. Arriving on the site at 8 a. m. it was delivering power to the grid at 6 p.m. on the same day.

Fig. 9 represents the small siphon of la Magnannerie on the Mayenne River; this design proves very economic for small plants and is particularly suited for modernizing old water mills.

When we discussed the overlap between Pelton and Francis Turbines, I remarked that the number of units is usually different when comparing a solution with Francis units or with Pelton ones. Here again when choosing between Kaplan and Bulb units for a low-head plant, the economics of the project is such that a greater number of Bulb units may be chosen than would be the case with Kaplans, the total cost of machinery and civil engineering being less. In some cases, it would not be fair to compare units of the same rating or same runner diameter as it would then be difficult to take full advantage of the possibilities of either Kaplan or Bulb units.

On some low-head power plants, the total economy on both machinery and civil engineering with Bulb units may amount to as much as 40 per cent as compared to the cost of the new conventional Kaplan power plant.

It is almost impossible to give general rules for the economics of such projects, especially concerning the civil engineering costs which will vary so much with local conditions. However, to give a general idea we can say, being aware of the limitations of such types of statement, that for heads above 20 m the conventional Kaplan still holds its own. For heads between 20 and 10 m we find the overlap between not only Kaplan and Bulb, but also with a few other types, such as pier-type power plants and shaft type units. For heads lower than 10 m Bulb units would be chosen. In fact, many of these hydroplants with heads lower than 10 m would not have been economic with conventional units. Bulb units have opened a new field for hydro-power. For instance, low-head navigation dams can now be equipped economically with Bulb unit hydroelectric plants.

TIDAL POWER TURBINES

Another field opened up now by this new type of turbine is that of tidal power. To be efficient, tidal power requires units which can be operated both ways as turbines and preferably also both ways as pumps. The Bulb unit lends itself very well to such a difficult type of operation and in various parts of the world tidal power plants are being contemplated, taking advantage of the new possibilities of these machines.

An example of a tidal power turbine is given in Fig. 10. This has been designed for the Rance estuary project and a full scale unit will be tested at Saint-Malo nearby. The runner diameter is 5.80 m. The rated horsepower is 9000 kW at a 5-m head. The Rance Power plant will in all likelinhood incorporate 40 such units.

The new possibilities offered by Bulb units for tidal power plants have encouraged a closer study of tidal power possibilities and many new cycles have been investigated. In actual fact, tidal power plant calculations are very complicated, as variations in turbine efficiency, alternator power limitations, the best choice of starting and finishing conditions, etc., have to be considered.

Nowadays, the only approach to this problem is by the use of electronic calculating machines, even if the utmost use is made of existing theory to guide the machine to the best result for any given tidal cycle and any desired cycle of power production. Of course, in practice, a succession of such cycles should be examined. For instance, a plant required for peak-load production must follow a solar cycle of output, whereas tides of course follow a lunar one. Hence, from day to day there is a change of phase. These calculations have not only shown that much more power could be developed than expected in the past but also that the power could be delivered to the grid with much more flexibility than was considered to be the case some years ago. For instance, the Rance power plant could be operated for peak-load production.

A few words can be said here about tidal cycles as the new improved cycles have a direct bearing on the design of the hydraulic turbines to be used.

Let us consider first a one-way simple tidal cycle where the reservoir is gradually filled through gates during the rise of the tide. The gates are then closed at high tide, and when the tide has gone down sufficiently the reservoir is emptied through the turbines.

With this simple operation the maximu enemrgy could be developed by waiting almost for low tide before emptying the reservoir through the turbines. This energy would then be developed in a very short time and the power plant would stay idle for the rest of the cycle. This, of course, is unpractical and uneconomic but can be taken as a yardstick of comparison for the energy developed by other cycles or modes of operation. This maximum energy E has sometimes been called the "wild energy" of the tide.

With the same simple operatoin of the reservoir, if a more even production of power is required with a smaller number of units, it will be necessary to start the turbines long before low tide so as to deliver power continuously for several hours. Instead of taking advantage of the maximum tide range the units will then operate at a somewhat smaller head perhaps only equal to half the tide range. Economic studies show then that the energy produced will be of the order of $\frac{1}{2}$ E.

Now, taking advantage of the fact that Bulb units can be operated as turbines with the water flowing either way, the same power plant could be used in filling the reservoir through the turbines, thereby producing more power. In such a cycle the so-called "wild energy" would be of the order of 1.8 E according to local conditions, and the economic design and operation of the plant would give an energy of the order of 0.9 E.

It may look surprising at first that in operating the units both ways, with efficiencies of the same order the energy is not just doubled. It should, however, be borne in mind that the reservoir changes in area with height and this implies quite different volumes of water for the various operations.

If now we take advantage of the fact that Bulb units can be operated either way as a pump, an improved cycle can be devised.

When the reservoir is full, at high tide, more water can be pumped into it against a small head. This added water will later on be passed through the turbines under a much greater head. This supposes that some energy can be drawn from the grid for pumping, the energy given back later being many times greater.

When the reservoir is emptied at low tide it can eventually be lowered further still by pumping so that more energy could be developed later on when filling again through the turbines.

At certain phases of operations gates would be used either to speed up filling or emptying of the reservoir.

In a cycle where advantage is taken of pumping facilities the total energy economically produced, once that taken for pumping is deducted, may amount to as much as 1.2E according to local topography of the reservoir.

This shows how important it is for tidal power units to be able to operate either way as pumps. These considerations affect of course on many details in the design of the machines.

These are not ordinary pumps either. They start operation against a zero head and are used only for very small heads. In that case the conventional efficiency coefficient loses its significance and the total pumping energy used during a cycle has to be compared with the volume of water pumped.

The Rance project now under study is designed for such pumping operation. Fig. 11 gives a cross-section of the future power plant.

CONCLUSION

In thisvery brief survey of hydraulic turbines in evolution some of the most striking advances in recent years have been reviewed. Much more could be said on the mechanical designs of the machines which have also improved steadily.

Contrary to some beliefs, it has been shown that very important progress has been made through the years. Various attempts made in the past to fix a ceiling to the possible performance off hydraulic turbines have utterly failed. Wherever enough research has been made it has been followed by progress, sometimes spectacular in nature.

Through my connection with hydraulic for more than 30 years, I have a feeling that more progress is to come in a not too distant future, on many aspects of hydraulic turbine design and operation. That is why the various curves given in this paper should be considered only as present trends.

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A Study on a Model Pelton Wheel

By R. K. K. R. GOVINDARAJAN
and
S. NAGARATNAM

Since the invention of the Pelton Wheel considerable work has been done to increase the efficiency with increasing capacity of an individual unit. Most of the improvements came from the laboratories attached to turbine manufacturing plants as refinment demanded empirical studies in the 'art' of turbine design. Certainly, to accept the present achievement as final will amount to falsifying the progress made during this century. Though the margin of gain will be comparatively less, yet possibilities for improvement exist. Hence nearly two years ago a programme was started to design and manufacture small model hydraulic turbine units with the help of the P.S.G. Industrial Institute. The present paper is a summary of the progress made so far and of the future plans.

HISTORY

It is natural that the time honoured water wheel should lead to the development of a reaction unit than an impulse unit, least of all to that of the Pelton Wheel which takes the middle position in the chronology of inventions of the three major types. In the year 1946, Escher Wyss in Zurich developed the first tangential wheel which led to the outcome of Girard impulse turbine in 1870. But it took another decade to utilise efficiently the impulse in a waterjet in the hand of L. A. Pelton of California, U.S.A. In his attempt to improve the efficiency of the then existing impulse unit of cup shaped buckets, he split the buckets, constructing different forms from oyster shells. From his models, an experimental unit was built at the University of California and that led to his patent in 1870 of an improved wheel with a square jet.

THEORY AND DESIGN

Abbreviations:

C_1	=	Mean velocity of jet
C_2	=	Exit velocity of jet from buckets
	=	Bucket outlet angle
d_o	=	Least diameter of jet
D	=	Pitch circle diameter
	=	Angular velocity
U_1	=	Peripheral velocity of pitch circle
	=	Mass Density
Q	=	Rate of flow
H	=	Total head
T	=	Torque
P	=	Total energy output.

The conversion of the energy in the jet occurs in the buckets (Fig.3). Torque is produced on the wheel by the force produced due to change of momentum of the impinging jet. Assuming steady flow.

$$T = \frac{PQ}{g} \quad \frac{D}{2} \quad (C_1\ C_2\ \text{Cos})$$

$$\text{and} \quad P = T = \frac{Q}{g} \quad \frac{U_1}{2} \quad (C_1\ C_2\ \text{Cos})$$

Bucket resistance, windage on the rotating wheel, and the inclination of the bucket to the jet at entry reduce the ratio U_1/C_1, from the theoretically best value of 1/2. Such reduction is a function of specific speed. The unit under study has been designed a specific speed of 3.76 to develop 4.0 horse power under a head of 120 feet of water, with an inflow of 150 gpm at 750 rpm. Depending upon the specific speed, U_1/C_1, d_o/D have been chosen as 0.46, 13.3 respectively.

Main considerations for the number of buckets in a disc are that buckets should be close enough for at least one bucket to be in full contact with the jet at a time and also buckets should be farther apart so that the exit jet does not interfere with the on coming bucket. These can be achieved by proper spacing of buckets and giving a small exit angle to the bucket. Also fewer buckets will produce less disturbance as the number of entries into the jet is less for a given speed. There are as many empirical formulae for the number of buckets as there are authors on the subject and hence it was decided to find out the best combination by carrying out actual tests on various numbers of buckets. It was decided to make three separate discs of 16, 18 and 20 buckets each, as most of the values of the empirical formulae fall in that range. The exit angle of the bucket is 15. The discharge loss will be less than 4% as the ratio of meridional component of the exit velocity to C_1 is less than 0.2.

For the shape of the bucket the following ratios have been used:

$$l/d_o = 2.5;\ b/d_o = 3;\ t/d_o = 0.86;\ D/d_o = 1.0.$$

$$\frac{D_1}{d_o} = 1.9; \quad \frac{S}{d_o} = 1.2$$

The structural parts have been designed giving consideration both to the runaway speed and the full jet under brake conditions.

Another important part in the hydraulic design is the nozzle opening and spear arrangement used for flow regulation. The approaching flow should be free from vorticies and the wetted area should be less to minimise frictional loss. Also nozzle and spear should be able to produce compact, round jet without any spray or cavitation at spear tip. Keeping the nozzle same, two types of spears have been tried.

Heavy type ball bearings have been used. The bearing near the runner has been protected from water by a water thrower and a suitable sealing arrangement. The unit has a usual rope break around a drum of 1'-6" diameter and the drum is water cooled.

RESULTS AND CONCLUSION

Performances curves of discs with 16, 18 and 20 buckets are presented in Fig. 4. Those tests were conducted with Spear No. 1. (Fig. 1) The disc with 20 buckets has definitely lower efficiency values. The disc with 18 buckets was chosen as the best as it has slightly higher efficiency at part loads than the disc with 16 buckets. In Fig. 6 is presented the characteristics of the wheel for .2,.4,.6,.8 and 'Full opening' conditions with spear No. 2. The position of the spear at designed discharge has been taken as 'full opening' position. The part opening is the ratio of the horizontal movement from the closed position of the spear between that part opening and the full opening. Such a procedure avoids ambiguity when spears of different travel lengths and shapes are tested with the same nozzle.

Fig. 1 and Fig. 2 are sketches of spear nos. 1 and 2 adopted from Gibson and Nechleba respectively. Spear having conical shape coverages the jet into cylindrical form faster and it has less travel length for complete closure of nozzle opening. Test results for both the spears under similar conditions indicate little difference.

IMPROVEMENTS PROPOSED

A comparison has been attempted in Fig. 5 between the Pelton Wheel available in the laboratory (B.H.P. = 1.72 at head-50 feet) and the P.S.G. Pelton Wheel. The Pelton Wheel available in the laboratory is a foreign made unit and has been maintained very well through the years and its performance for comparison is considered reasonable.

Due to higher percentage of mechanical and windage losses, it is natural that the overall efficiency of a laboratory unit is less than a large commercial unit where efficiency as high as 88% has been achieved. But the lower efficiency of the P.S.G. unit can only be explained that it requires further improvement. It is a known fact that more than half of the total loss occurs due to friction in the bucket. With increase in smoothness of bucket surface, we expect to increase the efficiency. And also we need study the approach conditions and performance of bucket. To further this study, it has been proposed to construct a special apparatus to find the efficiency of individual buckets under different nozzle and spear shapes. Already such an equipment is under construction and as soon as a series of test are completed and a better combination devised, we propose to incorporate it in the Pelton Wheel. It is hoped that such a systematic study will not only enhance the efficiency of the Pelton Wheel, but will add to the accumulated knowledge of design data.

Stoboscopic viewing indicated that the lowest bucket was not fully occupied by the jet as it gives out major portion of its energy. It has been, therefore, decided to increase the height of bucket and slightly change the curvature.

Acknowledgement

The idea to make a model Pelton Wheel is not new to the P.S.G. Colloge of Technology. But such an idea could not have come to fruition but for the inspiring guidance of Sri G. R. Damodaran, Principal and the wholehearted cooperation of Sri Doraiswamy of the P. S. G. Industrial Institute. Sri R. K. K. R. Govindarajan was responsible for design, production and for the close co-ordination between the workshop and the laboratory. He was assisted by Sri V. Krishnamurthi and Sri N. R. Palaniswamy in testing. Sri S. Nagaratnam was consulted at the testing stage and he was responsible for checking the design and the preparation of this paper.

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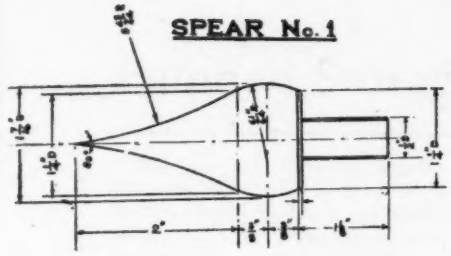
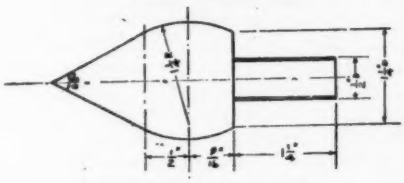
A Study on a Model Pelton Wheel

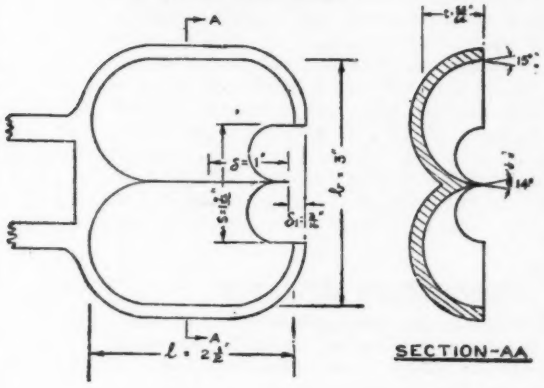
SPEAR No. 1 FIG. 1



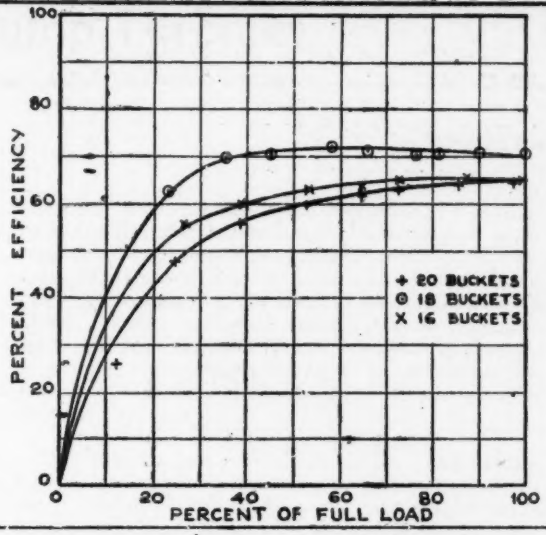
SPEAR No. 2 FIG. 2



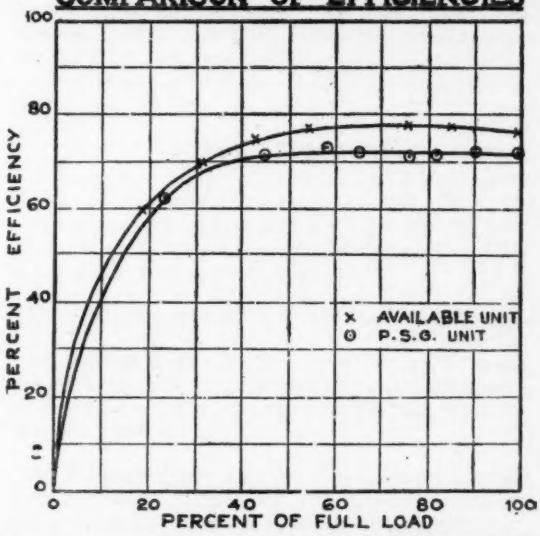
BUCKET DETAILS FIG. 3



RUNNERS WITH DIFFERENT NUMBER OF BUCKETS FIG. 4



COMPARISON OF EFFICIENCIES FIG. 5



CHARACTERISTICS OF THE PELTON WHEEL

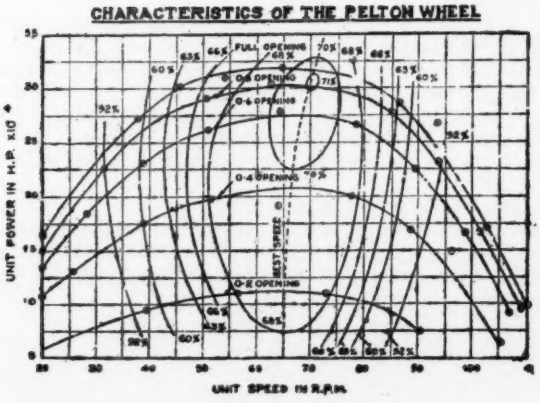


Fig. 6

Pumped Storages and Pump Turbines

By N. RAJARATNAM

The Electricity supply industry has the problem of ensuring high load factors. It is done in two ways. On the commercial side, it is tackled by selling current at special low rates during off peak hours and by applying two part tariffs which charge separately for maximum demand and current consumed. On the technical side, it is done either by storing the energy produced by the base load plant in the off-peak hours or by installing special peak load plants on which the fixed annual charges are low. One method of storing the energy is by pumping water into a high level storage reservoir at times when the general demand for power is low and by using the water thus stored to produce hydro-electric power during peak load periods. This is the general principle of the pumped storage schemes.

In the early stages of the development of pumped storages, separate pumps coupled to motors, mostly of the centrifugal type were used for pumping up the water. Later, the pumps were coupled directly to the electrical generating machine itself which was made to work as a motor in the pumping cycle. As a further development, reversible turbines were evolved which can work as a turbine for generating power and as a pump for pumping up water. These reversible hydraulic machines known as Pump Turbines can be designed either as a turbine or pump and as a single speed machine or as a machine with two designed speeds.

HISTORICAL DEVELOPMENT

The idea of pumping water into high level reservoirs is not new. The first pump for this purpose, of which records are available, was installed at Zurich in 1882. This was a reciprocating pump. The first centrifugal pump for this purpose was installed at Luino in Italy in 1894. It was of 75 h.p. capacity, pumping against a head of 210 ft. into an artificial reservoir and was used by a wind mill. Between that date and 1925, the size of the units increased slowly but steadily upto about 5000 h.p.; but no pump of exceptional size was designed until in 1928 when in the Niederwatha Station near Dresden, four 27000 h.p. pumps and 30,000 h.p. turbines were put into operation.

These units were the forerunner of a number of large capacity pumping sets built in Europe during the subsequent five years of which the largest in capacity at that time were the four 36000 h.p. pumps at the Herdecke Station in Ruhr. Recently a pump of 62,000 h.p. was put into operation at Providenze in Italy and units of 85000 h.p. have been completed at Limber in Austria.

The development of pumped storages has largely been confined to Europe and of a total of 850,000 h.p. of pumping capacity installed throughout the world in 1951, for these schemes about 600,000 h.p. was to be found in Germany alone and for the most part of the remainder, in order of the aggregate capacity between Italy, Switzerland, France, Austria and U.S.A. Isolated schemes exist in Sweden, Spain and Chile. In Great Britain, one small installation of 230 h.p., built in 1918 on the river Tweed was the only example of pumped storage until the North of Scotland Hydro-electric Board put in hand a scheme at Sror Mor near Inverary.

In South America several interesting pumped storage plants have been erected and it is here that the development of the single unit reversible pump turbine was first practically applied. Recently another pump turbine, claimed to be the 'World's largest Reversible Machine' has been installed at Hiwasse, TVA. The English Electric Company has designed a novel type of pump turbine with the 'Deriaz Runner' for the Ontario Hydro Electric Power Commission.

PUMPED STORAGES

A pumped storage hydro-electric plant may be defined as a plant in which the water that is used for power generation is pumped back into the high level reservoir using off-peak power, for generating additional power during peak periods.

Pumped storages may be divided into three groups:

(*a*) **Storage Type:** In this type, water is drawn from the reservoir to generator on-peak power and a portion of this water is pumped back in the same day or in the same week.

(*b*) **Pondage Type:** This is a closed circuit pumped storage system. All the water that is used for generating peak power is pumped back into the higher reservoir. In this system, the only quantity of water that is to be replaced is that lost by leakage and evaporation.

(*c*) **Part-Head Type:** In this type, the water that is available at a moderate head is pumped to a greater height to generate more power.

Pumped storages may be designed for one of the following three main objects.

(*a*) For improving the daily output of a particular station. Most of such installations are small and are used in conjunction with low head run-off river hydro stations where it is either impractical to store the water

during the hours of low demand or it is necessary to maintain a uniform river discharge.

(*b*) For improving the daily operating conditions of a large inter-connected group of power stations, thermal or hydro-electric water is withdrawn from the upper reservoir to produce peak power and is replaced by an equal quantity in the same day or week using off-peak power from one of the group of stations.

(*c*) For supplementing the natural inflow into a storage reservoir which is insufficiently supplied by its catchment area or for increasing the firm output obtainable with the storage capacity provided.

ECONOMICS OF PUMPED STORAGES

The economic feasibility of any such scheme will be largely affected by the source of power for driving the pump. This may be surplus hydro power or in a mixed hydro and thermal system, it may be supplied from steam power stations.

If the power for pumping is derived from run-off river plants or from unsalable off-peak energy, there is no extra cost to be paid for this energy and only the fixed charges on the pumping equipment will have to be taken into account. In an inter-connected system of hydro-electric power stations, power for pumping in one station can be obtained on a loan basis from the off-peak energy of another station which can be replaced by the off-peak power at some other part of the day or week. But if thermal power is used for pumping, it may increase the working cost of the scheme since the efficiency of conversion is likely to be low.

Further, pumping of the water can be done either in the same day or in the same week. In weekly pumping system, the reservoir becomes empty on Friday night and is refilled during Friday, Saturday and Sunday nights. In this case, the reservoir capacity should be more.

The advantages of the pumped storage system are:

(1) It is comparatively independent of the rainfall and drainage area.

(2) In more settled areas, the cost of the land and other damages are apt to cost less for a given reservoir capacity because the plant is on a tributary instead of in the main river.

(3) Since for a given reservoir capacity, the drainage area is less, the provision for floods is less expensive.

(4) The plant that pumps from a large river can be used as a reserve plant because water released in an emergency can be stored by pumping.

(5) Finally among the most important advantages is the low cost of adding pumping and power equipment to an existing development to obtain more firm or reserve power.

PUMPING EQUIPMENT AND EVOLUTION OF PUMP TURBINE

The plant for a pumped storage scheme consists of normal turbo-generators with some type of pump. In the first pumped storage plant, a reciprocal pump was used for pumping up the water. Centrifugal pumps came into the picture and with the advances made in the design of hydraulic machinery, pumps of very high capacity (60,000 h.p.) have been installed in some of the recent plants. In the next stage, the principle of using the turbine itself for pumping the water was conceived and thus the reversible machine known as the pump turbine came into existence. Little attention was given to the reversible machine until the late 1940s when considerable research was carried out in U. S. A. with models of turbines for higher heads. This led to the production of several prototype installations, the most notable of which is at Hiwasse dam of the TVA. A recent British development is a novel type of runner which is of the Francis type with movable blades. This was developed by Paul Deriaz at the English Electric for Sir Adam Beck Station in Ontario.

GENERAL PRINCIPLES OF DESIGN

These pump turbine units are not generally intended for regulating load but for supplying blocks of prime power to the system. Therefore, it will have a relatively slow acting governor or no governor at all, and even at the higher heads there wil not be much pressure rise in case the load is suddenly lost. Therefore a pressure relief valve will not be required and problems of water hammer will not be involved. In such cases, a unit may reach its maximum overspeed (runaway), but this is much lower than in a conventional Francis turbine and the machine is built to operate safely for the short duration of these infrequent overspeeds and can be shut down in the usual manner.

Since the pump turbine has to work both as a turbine and pump, the design of the runner must of necessity by a compromise between the best for a turbine and that for a pump. As the specific speed of a pump is greater than that of a geometrically similar turbine, two different speeds are necessary if the maximum efficiency is to be obtained in both directions of rotation. Alternatively, if a single speed is adopted for both operations, this speed must be a compromise between the best for the turbine and that for the pump. It has been found out that for pump turbines operating at heads below 150 ft., operation of the machine at the same speed in both the cycles will produce optimum results. However, for heads above 150 ft., there is a considerable increase in the turbine output and better overall efficiency if the turbine can be operated at a lower speed than is used for pumping as shown in Fig. 1. If the machine is to be designed for two speeds, the generator/motor and its switchgear will be more complicated and expensive. Whilst the Baldeny machine is of the two speed type, it is of only 1100 kW and is of the induction type with two separate stator windings. Messrs. Allis-Chalmers Manufacturing Company have built a synchronous generator for a reversible machine which will operate

at 300 r.p.m. as a motor and 257 r.p.m. as a generator. The performance curve of a pump turbine is given in Fig. 2.

RUNNER FOR AXIAL FLOW PUMP TURBINE

The runner of an axial flow reversible machine can be designed in a way similar to that of the axial flow turbine. But the problem of reversibility, i.e., the reversal of the direction of flow, when the turbine works like a pump should be carefully studied; since the usually sharp trailing edge becomes the leading edge and the well rounded leading edge the trailing one. As a result, there are two serious disadvantages. Firstly, there is a considerable drop in efficiency. Secondly, due to the flow from the sharp edge to the rounded edge, there is an unfavourable pressure distribution in the hydrofoils leading to the danger of cavitation.

Brown has shown that in the case of thin aerofoils, the lift slope curve and drag remain unaltered when the flow is reversed both in sub-sonic and super-sonic flow, provided the trailing edge velocities are finite and suction forces are negligible. But in the case of axial flow machines, the blades can in no case be considered as thin sections. The sections at the blade tip or nearby can be regarded as thin while at the root or nearby fairly thicker sections are present. On strength considerations root sections have to be fairly thicker and further the usual design assumption of constant angular momentum demands the adoption of maximum lift at the root which can properly be provided by thick aerofoil sections. This problem of reversibility was studied by Kar.

Investigations were made with elliptic sections. This section has the advantage that due to symmetry, the aerodynamic characteristics remain almost unaltered on the reversal of flow. Also the pressure distribution is favourable and therefore quite acceptable on cavitational conditions. Reason against the adoption of a elliptic section for having relatively higher value of E ($= \frac{C_D}{C_L}$, C_D being the coefficient of drag and C_L the coefficient of lift) than a conventionally used aerofoil section and thereby affording less profile efficiency is not fully justified. Williams and Brown have shown that the values of E are comparable at high Reynolds numbers with those of conventional profiles. This arises from the fact that the influence of the Reynolds Number on the aerodynamic characteristics of an elliptic section is much more pronounced As a result, since the axial flow machines operate at higher Reynolds Numbers good and even comparable efficiencies can be obtained by using elliptic sections. Model tests showed about 7% decrease in maximum efficiency by having elliptic sections rather than the conventional ones. Of course the adoption of elliptic sections has its own limitations inasmuch as the favourable values of E occur only at higher incidences and unlike the usual sections, which retain the characteristics over a considerable range, it has a poor performance at other values. Besides, the advantage of favourable pressure distribution is lost at incidences other than the designed one.

RUNNER FOR THE FRANCIS TYPE PUMP TURBINE

The runner for the normal pump turbine is designed in a way similar to that of the Francis turbine. In the Hiwasse machine, the world's largest reversible pump turbine, the diameter of the Francis runner is 266 inches. A ring of 20 adjustable wicket gates controls the flow of water to the unit's impeller runner for power generation and likewise regulates the flow of pumped water for greatest efficiency. When starting as a pump, the wicket gates the wicket gates are closed and a compressed air blow down system dewaters the runner. This reduces the motor starting torque requirements and eliminates system disturbances. Reduced voltage starting limits current on rush below normal load. The motor is thus started and brought upto speed. Release of air pressure allows the water to rise in the impeller, the wicket gates slowly open to the proper setting and water flows up the penstock into the storage reservoir. Much of the sequence in the reverse is followed at the end of the pumping cycle. For power generation, the unit is started in the same manner as a conventional hydraulic turbine unit.

In Ontario's major hydro-electric development, at Sir Adam Becks Station, six reversible feathering blade—Francis type pump turbines developed by Paul Deriaz at the English Electric, have been erected. The runner is of the diagonal flow type with eight movable trunnion supported blades sloping downstream at an angle of 45 degrees from the runner hub. The blade angle is adjusted by a vane type servomotor located in the runner hub. The hub and inner edges of the blade are finished spherical permitting unrestricted feathering of the blades and also close clearance tolerances. Likewise the outer edges of the blades and throat ringwall are spherical. Because of the close clearances between the hub, the blade edges and the throat ring, the blades in the closed position form a reasonably water tight closure and conventional wicket gates and their operating mechanism have been eliminated. When the machine is shut down, additional thrust may be introduced on the top of the runner hub by closing the cover drain valve which produces a deflection and reduces the clearance between the ends of the blade and the throat ring. This reduces blade gap leakage to a minimum. The steel plate casing is set well above the centre of the blades to ensure axial flow past the blades.

The unit when acting as a pump is started with the blades in the closed position and brought upto speed without the necessity of depressing the water level in the draft tube. This requires very little effort other than inertia of the runner and motor and starting will consequently be across the line with the expected onrush current of about 25 percent of the running current. When synchronous speed has been attained, the blades will be gradually opened and pumping begins.

Another pump turbine designed in Germany embodies a Kaplan type turbine with fixed guide vanes

and automatically adjustable runner blades. The design of the pump turbine runner is under constant development and improvement.

SPIRAL CASING

To get a higher efficiency in both the systems of operation, a completely revised form of spiral casing and diffuser was developed by Deriaz. The various stages of development are shown in Figs. 3, 4 & 5. It will be noted that the final design has a large oblique diffuser and the guide vanes have been dispensed with.

THRUST BEARING

The pivoted shoe type thrust bearing is ideally suited for the reversibe machines. In the Hiwasse Machine, it takes a load of 1,365,000 lbs. These thrust bearings are fitted with high pressure lubrication to reduce the breakaway torque in starting both for motoring and generating.

DETAILS OF FEW PUMP TURBINES

Pump turbine at Hiwasse—TVA

It was erected by Allis Chalmers Manufacturing Company. Electrically, it is rated as 70,000 kVA; as a hydraulic generator; and 102,000 h.p. as a pump driving motor.

Power factor is 95%—13,500 V—135.9 r.p.m. as a Motor and 85% Power factor—105.9 r.p.m. as a generator.

Diameter of the Francis type runner is 266 inches—Finished rotor weight is 300 tons—load on thrust bearing is 1,365,000 lbs. Turbine to generator 80,000 h.p. under 190 ft. and pump to pump up 3900 cfs. against 205 ft.

Pump Turbine at Sir Adam Beck Station

Pump turbine—erected by English Electric Company.

Operating as pumps.
Discharge — 4000 to 5000 cfs
Head range — 90 to 60 ft.

Operating as Turbines.
Power generated — 40,500 h.p.
Head — 80 ft. (range 80—30 ft.)
Discharge — 5600 to 4100 cfs..

The units will be operated at 13.8 kV with transformation directly to 230 kV.

Acknowledgement

The author is deeply indebted to Prof. N.S. Govinda Rao, for his enlightening guidance in the preparation of this paper, expresses his very sincere thanks to Sri. K. Seetharamiah, Asst. Professor, for his valuable criticism and keen interest in the preparation of this paper.

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Pumped Storages and Pump Turbines

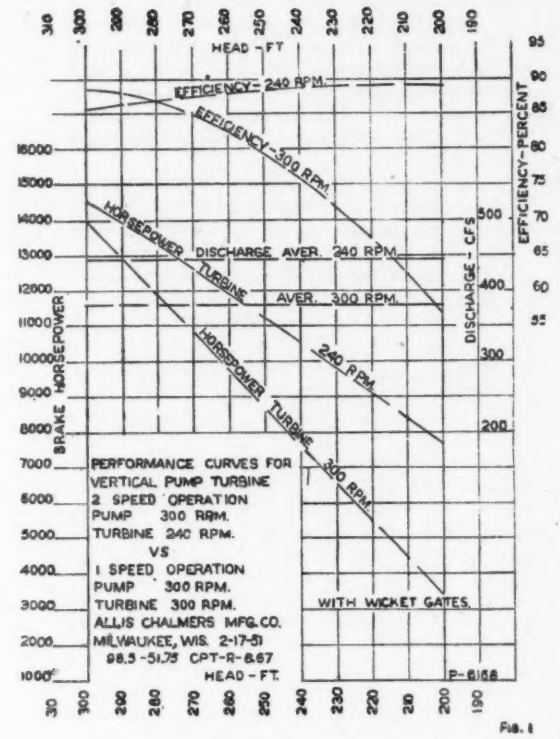


Fig. 1

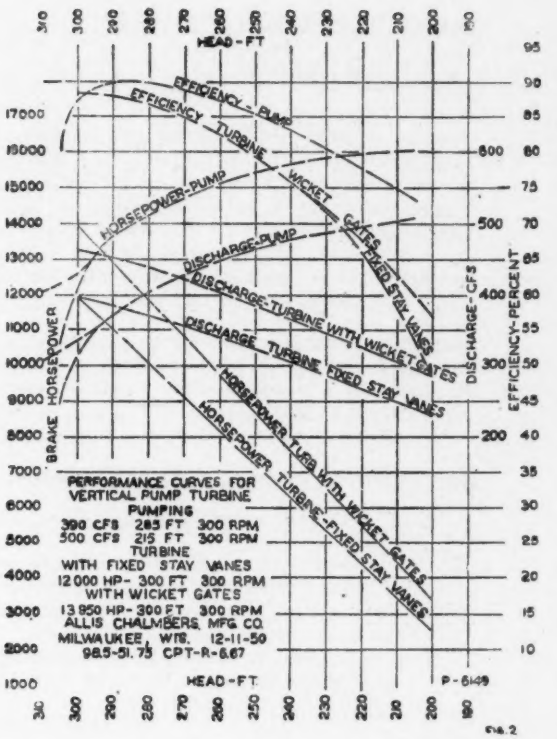


Fig. 2

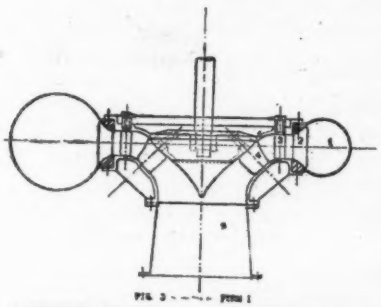


Fig. 3 - Form I

1. Conventional spiral casing
2. Stay vanes
   Gate apparatus
4. Variable pitch mixed flow runner vanes
5. Draft tube.

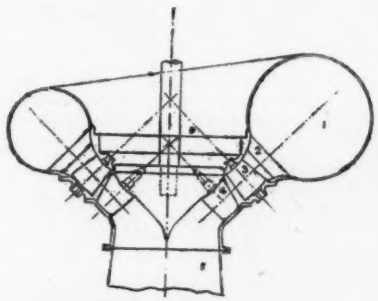


Fig. 4 - Form II

1. Spiral casing oblique design
2. Stay vanes
3. Adjustable gate apparatus
4. Variable pitch runner vanes
5. Draft tube.

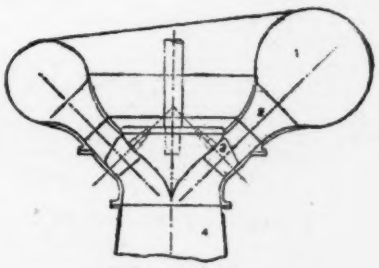


Fig. 5 - Form II

1. Spiral casing oblique design
2. Fixed diffuser vanes
3. Variable pitch runner vanes
4. Suction pipe.

Tidal Power and Tubular Turbine

By G. SURYANARAYANA IYER
and
A. S. NAGARAJA

The ebb and flow of tides are quite a familiar phenomenon for a sea-shore dweller. Tides are, as it were, the breathing of the sea, the pulse of the cosmos, which gives life to the oceans of the world and sets them in motion, and since the remotest ages, this mystery has haunted the mind of man and stimulated his heart and brain to meditate and search.

The cosmic potential of tides is so great that the energy that is harnessed upto the present from water appears trivial. A rough estimate of the total tidal potential of the world for a mean tide height of 0.7 m. is about 54,000 million h.p., in comparison to about 7,600 million h.p. the possible output of the continental water resources.

(tide height $=0.7$ m; velocity $=0.8\sqrt{2gH} \simeq 15$ f.p.s. Discharge $=204 \times 109$ cusecs, coastal length $=$ 25,600 miles.)

Attempts to utilise tidal power and kinetic energy offered by the perennial rythm of ebb and tide dates back as far as 11th century in the form of tide mills for grinding corn or gypsum or sawing timber. Upon the advent of steam engine, tide mills became unpopular until their importance re-emerged in the 20th century. A few years after World War I, schemes were thought of for the 'Severn' and two tidal plants in the vicinity of Liverpool in U.K. Similarly the untapped resource was duly recognised in France. Subsequently these tidal schemes were given up as it was thought they were uneconomical. The historic survey is a proof that some significance was given to tidal power which suggests itself as a suitable solution to the intensive search for new sources of energy at present. It is expected that within the next 10 years, the present needs of the world will be doubled.

WHY TIDAL POWER

(*a*) With the numerous harnessable rivers already developed to the full, and the increasing scarcity of convenient sites for setting up further plants,

(*b*) With the thermal plants, paying now for coal four times as much as in 1939 and hardly able to improve their efficiency, and

(*c*) With the economical production of atomic energy still being far distant, conditions are favouring more and more towards tidal plants.

Since the cause of ebb and flow is of astronomical nature, their progress and consequently the possible output of tidal plants can be computed years in advance with great accuracy.

A closer investigation of the possibility of harnessing tidal power reveals multiplicity of overlapping problems. Traditional hydro-electric practice applies only partly to tidal plants in the working of which cosmic forces intervene to a great extent. Therefore, we must realise that no practical solution can be arrived at without a close co-operation of the astronomer with the hydro-grapher and of the hydraulic engineer with the mechanical and electrical engineer, not to mention the economist.

To what extent can we harness tidal energy? Since the velocities of tidal flow are always very small, the utilization of high kinetic energy of tides would be hardly worthwhile, the investigation of the exploitation of potential energy only need be considered.

TYPES OF TIDAL PLANTS AND THEIR OPERATION

The types of tidal development may be

(*i*) One-way or Two-way single basin.
(*ii*) Single-basin with auxilliary basin.
(*iii*) Two basins operating together or separately
(*iv*) Three basins with intermediate chamber.
(*v*) Defour's three basin system.
(*vi*) Two-way operating single basin.

Methods of Operation

Single Effect: In the most simple type of tidal power scheme, a barrage enclosing an estuary or a bay creates a single basin. The potential energy of water in the basin is utilised when the basin is emptied, and this is known as the single effect emptying cycle (vide fig. 1).

There are three distinct operating phases:

(*i*) A filling phase:—The turbines are stopped and valves are open. With the tide rising water flows into the basin through the valves.

(*ii*) A waiting phase :—The turbines are stopped, valves are closed and sea-level falls.

(*iii*) A production phase:—The turbines produce energy by reason of the head created by the difference in level between the basin and sea.

In this arrangement, power is developed during the flow of water from basin to sea. A similar type of operation is when the turbines operate during the filling cycle. Since the estuaries are of irrigular shape, the volume of water moved for the same difference of

level will be more in the upper layers than in lower layers, the filling type of operation gives less energy than the emptying cycle. For example for Rance project, the ratio is about 2/3: 1.

Double Effect: In this method, energy is produced both during the filling and emptying of the basin, (Fig. 2). Before the advent of modern feathering blade turbine which operate in either direction of flow, it was necessary that arrangements should be made to ensure the flow through the turbines unidirectional.

Pumping: To improve still further the operation of double effect cycle, the modern turbines can be made to operate as pumps at slack water periods and the utilisable volume of water and the operating head is thus increased. The net gain in power production, inspite of power consumed by pumping is substantial and a further economic advantage is secured by the differential in value which exists between peak-load and off-peak energy.

TIDAL POTENTIAL

One-way Operating Tidal Plant:

Consider a basin operated from sea by a dam, the water level in the basin following the rise and fall of the sea.

Let H= tide height, V_B =capacity of basin, at corresponding water level H, $S_{(z)}$ =area of the basin corresponding to water level, z, in the basin.

The work performed during emptynig period will be

$$E_{B1} = \int_{Z=0}^{H} \gamma S_{(z)} Z \, dz. \qquad .. \qquad (1)$$

and during filling cycle,

$$E_{B2} = \int_{Z=0}^{H} \gamma S_{(z)} H - Z) \, dz. \qquad .. \qquad (2)$$

Thus the total tidal energy during a tidal cycle is

$$E_B = E_{B1} + E_{B2} = \gamma H \int_{Z=0}^{H} S_{(z)} \, dz = \gamma H V_B \qquad (3)$$

The above is the ideal value, but actually a fraction of it can be utilised,

$$\text{i. e.,} \quad E_B = \delta \gamma H V_B \text{ where } \delta < 1 \qquad .. \qquad (4)$$

With the help of artificial means, it is possible to arrive at a much higher degree of utilisation. As soon as the tide reaches the highest point, sluice gates in the dam are opened to admit water at high velocity to the empty basin and subsequently closed as soon as the basin is filled. Taking into consideration the surges occuring in the basin while being filled, tests models have established that a storage level of 1.5. H can be easily attained.

As there is no energy generation during filling period, the possible energy output can only relate to the emptying period, i.e., the ebb half of a tidal cycle, hence

$$E_B = \tfrac{1}{2} \gamma S (1.5H)^2 = 1.125 \gamma H V_B \qquad .. \qquad (5)$$

Therefore in the most favourable instance by using surges to the full, the energy production can be more than that would be possible normally without surges.

The above consideration apply exclusively to the plants with turbines only and, as aforesaid, utilisation possibilities could be substantially modified by adding a pumping system. If for instance, for a one-way plant operating with turbines and pumps, we may have in addition to main basin, an auxilliary basin, $F_{(z)}$ ($z \geq H$), with its bottom lying at the level of the tide height H. While the incoming tide fills the main basin the pumps coupled to the turbines lift water to the auxilliary basin to the level B. If the turbines and pumps are of equal capacity then we have

$$\int_{Z=0}^{H} \gamma S_{(z)} (H-z) \, dz = \int_{H}^{B} \gamma F_{(z)} (z-H) \, dz \qquad (6)$$

(i.e., energy produced during filling cycle is utilised for pumping to level B),
and if the pumped volume is V_p, then.

$$E_B = \int_{Z=0}^{H} \gamma S_{(z)} Z \, dz + \int_{2=H}^{B} \gamma F_{(z)} Z \, dz$$

$$= \gamma H (V_P + V_B) \qquad .. \qquad (7)$$

Comparing equations (2) and (7), we notice that the theoretical energy produced during the ebb period is increased by H V_p on account of pumping

Two-way Operating Plant :

In this system, a basin of appropriate size separated from the sea by a dam fills and empties solely through the power house. In contrast to other systems, no additional sluice gates or intermediate dams are required.

Once the high tide begins to fall and the level of the water in the basin stands above sea-level, condition a, (Fig. 3) the turbines are started as soon as required head is available (condition t) and pass the water of the basin down to a minimum level. By adjusting the guide and runner vanes, the turbines run idle and water continues to flow out of the basin (condition V). Subsequently the turbines operate as a pump, (condition p), the water level in the basin sinking below sea level, following a waiting period (condition a) during which direction of flow is reversed and the flood tide starts afresh. The turbines operating under load from the sea side (condition t') utilises

the head available. After a period of idle running (condition V′) and the subsequent pumping (p′) and a waiting period (*a*), the working cycle begins again.

In such a plant operating with double admission turbines and two-way pumping, and in the general case of a single basin without higher auxilliary basin, the pumps drawing their power from the network, we have the following four phases:

(*i*) The water level in the basin = low water level, being further lowered by—B by pumping, the pumps will draw from the network

$$E_{P_1} = \int_{Z=0}^{-B} \gamma S_{(z)} Z \quad .. \quad (8)$$

(*ii*) The rising tide fills the basin and the energy output of the turbine is

$$E_{t_1} = \int_{Z=-B}^{H} \gamma S_v (H-Z)\, dz \quad .. \quad (9)$$

(*iii*) Upon high water level being reached, the pumps raise the water level in the basin to C with an energy consumption of

$$E_{P_2} = \int_{Z=-B}^{C} \gamma S_{(z)} (Z-H)\, dz \quad .. \quad (10)$$

(*iv*) The turbines utilise the water in the basin down to the level zero, during which period, energy output is

$$E_{t_2} = \int_{Z=c}^{t} \gamma S_{(z)} Z\, dz \quad .. \quad (11)$$

Therefore, net energy produced in an operating cycle

$$E_B = E_{t_1} + E_{t_2} - E_{p_1} - E_{p_2}$$
$$= \gamma H \int_{Z=-B}^{C} S_{(z)}\, dz = \gamma H (V_B + V_{p_1} + V_{p_2}) \quad (12)$$

Therefore if turbines and pumps operate at both flow and ebb periods, the energy gain to be expected amounts to γ H $(V_{p_1} + V_{p_2})$ as against turbines exclusively in operation and γ H V_{p_2} in the case of one way pumps delivering into a high auxilliary basin.

The judicious selection of the sea-dam site in a bay or estuary makes it possible by taking advantage of resonance to increase the tide range during the tidal cycles since each basin has a natural frequency of its own depending on its size. In a basin of length L_m, with a rectangular C.s. and a depth t, the natural frequency is $T = 2L/\sqrt{gt}$. When the natural frequency approaches the frequency of the tide, the tidal impulses which set the water in the basin into oscillation are reinforced. There is then the possibility, as proved by computation, of tide range increasing upto 10%.

TURBINES FOR TIDAL PLANTS

The tidal power development has been rendered possible by one vital factor—the development of a horizontal Kaplan type turbine—alternator unit suited for very low heads and capable of operating either as a turbine or a pump with the water flowing in either direction. Such a unit offers the real solution to the problem of the intermittent character of the tides and this, coupled with the fact that the savings in the capital cost both of the units themselves and of the associated civil engineering construction works are so substantial, has transformed the economics of the tidal power with the knowledge of aerodynamics, it proved possible especially with high speed turbines to increase efficiency by adopting favourable blade profiles and by reducing flow losses in the inlet chamber and draft tube.

In case of axial runner wheels employing spiral casing and elbow shaped draft tube, the flow from inlet to runner wheel is relatively long and the admission to the runner wheel and the subsequent discharge in an elbow shaped draft tube involves repeated deflection of flow. Therefore the type of turbines generally used for run-of-river plants are not fully suitable for tidal duties.

DEVELOPMENT OF TUBULAR TURBINES

An arrangement in which a horizontal axial flow turbine drives a vertical generator through a bevel gear installed inside the guide wheel-hub proves more favourable in meeting the requirements of good water flow and economical expenditure on building work. A pre-requisite for the adoption of this design is reliable sealing of the gear which operates in an oil bath. If this can be fulfilled, then the generator can also be installed within the guidewheel hub. Supply of cooling air and discharge of possible leakage water can be accomplished by employing hollow wicket gates. It is here that the development of axial flow tubular turbines, also known as "Arno-Fischer" turbine commences. At the runner end, the guide-wheel hub is sealed against rotating shaft by a stuffing box employing rubber lips. For shutting off the turbine, there is a butterfly valve on the intake side and a stop-log on the discharge side (Fig. 4). The tubular turbine has the advantage of favourable flow both at entry and exit resulting in better discharging condition.

The experience gained at the Rostin plant, Pomerania, shows that in principle, it is possible for atleast small outputs, to arrange the generator inside the guide-wheel hub provided the external dimensions have not been chosen too small and the profile of the guide vanes are dimensioned enough to allow of sufficient air cooling in the generator.

The tubular turbine may be said to have originated by the late American Engineer L. F. Harza, who proposed a horizontal runner surrounded by a ring

which carried the alternator poles, the alternator thus forming an annulus around the water passage (Fig. 5). This design is specially suitable for large units. In this design, the generator again becomes easily accessible. With the large magnet ring diameter, it is easier to accommodate the required fly wheel effect for proper regulation of large units. At the same time, the guide wheel hub can also be favourably designed for the flow.

The main problem in this design is the sealing of the external rim of the runner wheel rotating at more than 20 m/sec., and the stationary tubular parts in such a manner that no water could come in contact with the windings of the generator. This led to various designs providing other locations for the alternator. Two solutions in particular proved significant. The first was to install the alternator in an open pit and the second was to construct a combined turbine-alternator unit in which the alternator was housed in a bulb completely surrounded by water.

The pit type represents the advantage of using a standard alternator which is permanently accessible for inspection. On the other hand, the discharge passage bifurcates and passes round either side of the pit and in the case of large capacity machines at very low head, the necessary size of the alternator tends to make this bifurcation excessively divergent, adversely affecting the hydraulic conditions. A possible solution to this difficulty is use of a speed-increasing gear between the turbine and the alternator to reduce the size of the alternator.

In the bulb-type installation, an adjustible two wheel turbine-pump, also operating in either direction of flow as a turbine or a pump is used (Fig. 6). The turbine pump and its hydraulically coupled generator are enclosed in a cylindrical tube widening at one end to a bulb shaped section and designed in accordance with wind tunnel tests. The respective duties of guide vane and runner are fulfilled either by the front or rear wheel according to the direction of flow. These have adjustible and reversible vanes and can be thrown in or out independently of each other. At a given position of the vanes, it is even possible to stop the flow through the turbine altogether. Sealing between runner rim rotating in water and the stationary turbine casing is effected by what are called lip seals. The lips seal so tightly that each turbine has a leakage of 2 to 4 litres/sec. which can be readily dealt by leakage tray and then led on to the leakage sump. This arrangement definitely avoids any increase in overall lengths of the turbine, but great care is necessary to ensure the presence of such a large body in this sensitive zone does not affect the hydraulic performance. On account of the fact that these are low-speed machines, one of the limiting factors in this design is the size of the generator. A successful suggestion to this is to introduce a planetory speed-increasing gear between the turbine and the generator. The need to reach the generator for maintenance led to an investigation of the hydraulic effects of introducing a vertical access shaft and the important discovery was made that the energy loss was only slightly greater than that due to normal supporting vanes. This led to the provision of another supporting shaft below the generator on which the entire unit can be installed. A comparison between the tubular unit of this design and of equivalent vertical Kaplan units for low heads shows an improvement in favour of tubular turbine, of 3 to 6% accompanied by reduced cavitation troubles.

Preference should be given to inclined designs in view of the still better inlet passage, the shorter inlet distance and the easier location of the access passage under the machine with the same or even smaller excavation depths. The tubular turbine is chiefly suitable for heads ranging from 6 to 15 meters. For lower heads, deep excavation will be necessary.

The types of turbine generally used in run-of-river plants are not generally suitable for tidal duties. The turbine shaft can be either horizontal as in tubular turbine or vertical. Tidal plants equipped with vertical plants and operating with one-way power flow only, do not substantially differ from the normal type of inland low head run-of-river installations, but if two-way working is to be made possible, the turbine inlets and draft tube must be gated to suit the direction of flow. Another solution would consist in providing a revolving elbow type draft tube which will not only create favourable flow conditions in either direction, but also take the place of the gate valves.

The Voith type of tubular turbine, with two sets of guide vanes and a single runner, would enable the layout of the power-house to be simplified to some extent: the runner being encircled by the generator, the widths from the inlet intake to the outlet could be reduced.

NEW APPROACH TO TIDAL POWER

The approach is based on a new type of water wheel called "Impact Water Wheel" (I.W.W.) conchieved by Percy. H. Thomas while researching on wind turbine. The performance characteristics of I.W.W. permit operation interchangeability as a pump or a turbine and also operation with the flow in either direction with the change of blade pitch, all with the same direction of rotation. Further it operates normally at constant speed even when the hydraulic head drops to very low value, of course, with a lowering of output and efficiency.

The I. W. W. resembles the airplane propeller, the wind turbine and the hydraulic ram as all these have intermittent impact features. In physical form, the I.W.W. closely resembles the wheel of the Kaplan turbine taken by itself omitting the scroll case and wicket gates. The I.W.W. has an enlarged hub and two or three narrow blades and is located at the narrow throat in the stream line flow channel between the approach and the draft tube. The water stream would be a spouting jet through the throat except for the rapidly rotating blades, which, working through the inertia of the water, force the flow to take the velocity, corresponding to the average pitch. Since the blades hold back the flow, they carry the whole

Tidal Power and Tubular Turbine

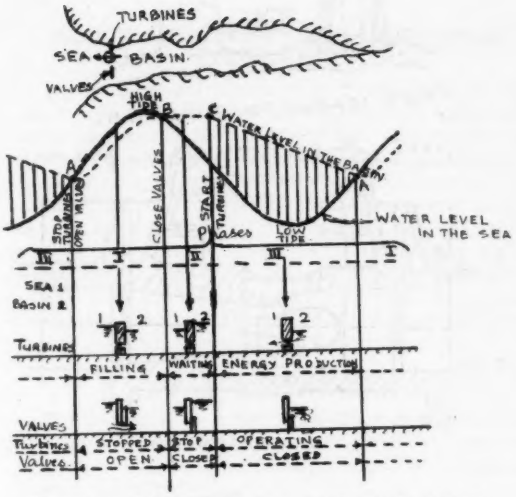


FIG 1. SINGE EFFECT EMPTYING CYCLE.

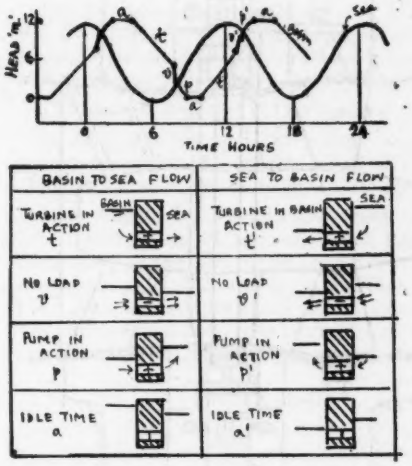


FIG 3. CYCLE DIAGRAM FOR A TWO-WAY SINGLE BASIN

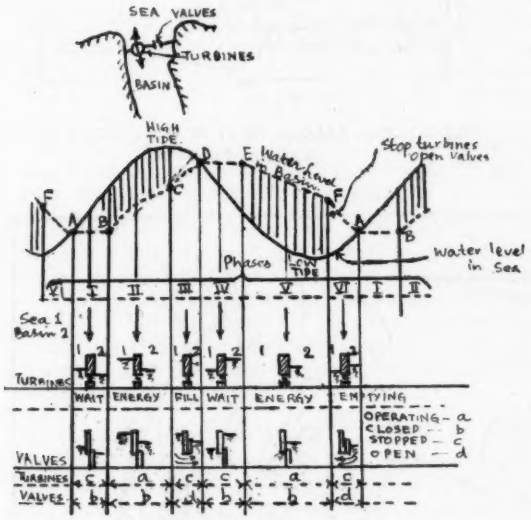


Fig. 2. Double effect cycle.

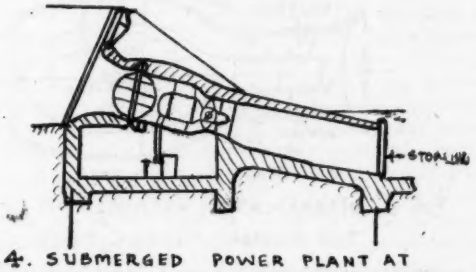
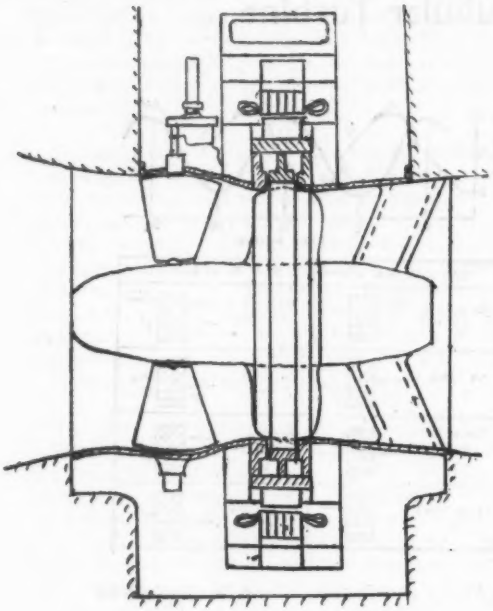


FIG. 4. SUBMERGED POWER PLANT AT ROSTIN WITH TUBULAR TURBINE.

FIG. 5. MAGNET-RING SHRUNK ON THE RUNNER



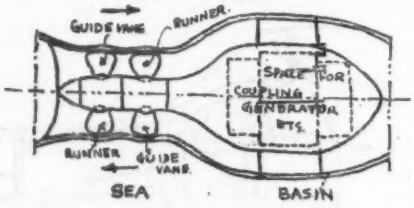


FIG. 6. DIAGRAMATIC ARRANGEMENT OF TWO RUNNER TURBINE-POMP

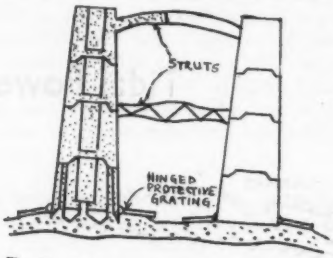
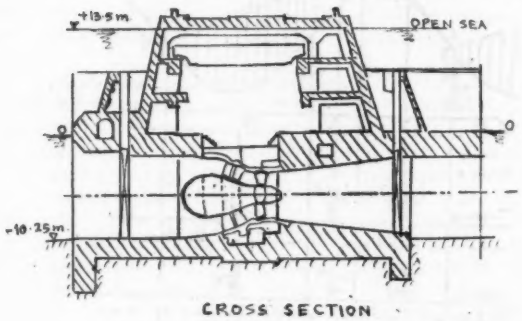


FIG 7. SEA-DAM FOR TIDAL PLANT



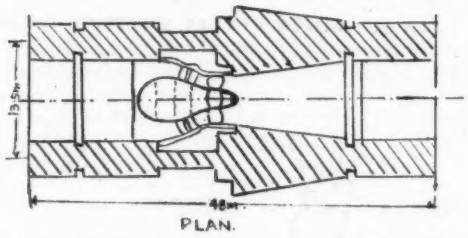


FIG 8. GENERAL ARRANGEMENT OF GENERATING UNITS. (RANCE PROJECT)

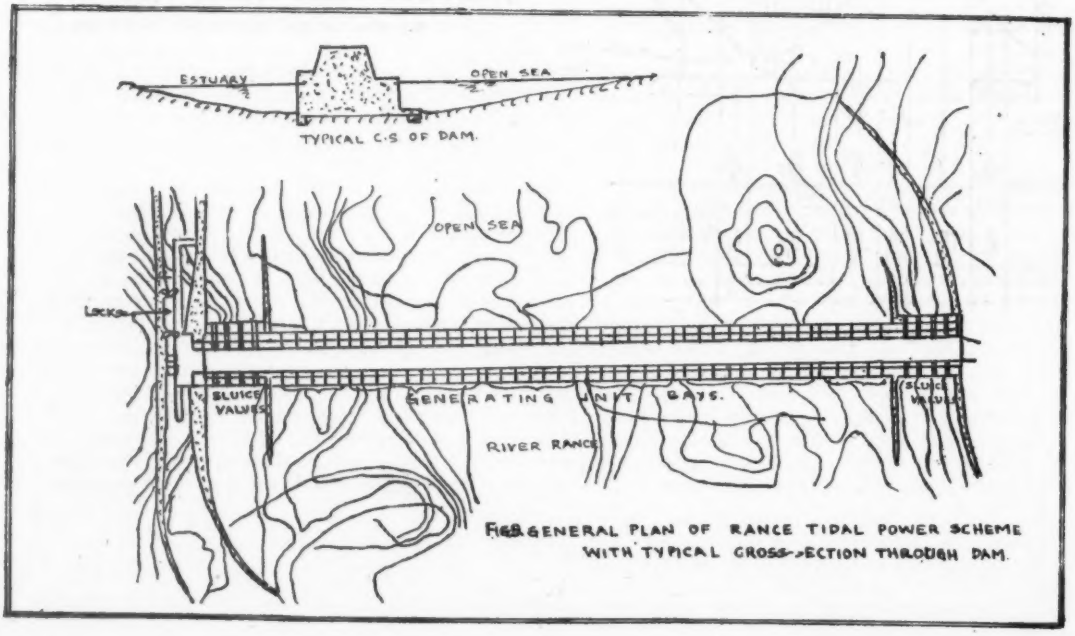


FIG 9. GENERAL PLAN OF RANCE TIDAL POWER SCHEME WITH TYPICAL CROSS-SECTION THROUGH DAM.

penstock pressure, and, since they have a pitch, there is a component of this pressure in the direction of blade rotation which constitutes the torque force on the wheel. This action of I.W.W. has much in common with ship's screw. The rotational speed of the I.W.W. is two or three times that of Kaplan runner, a character that is very favourable to the generator especially with low heads.

DESIGN ASPECTS OF CIVIL ENGINEERING STRUCTURES

The problem of foundation is of utmost importance in a tidal plant. The foundation of power house must be strong enough to withstand the catastrophic conditions of nature like storm. As a rule, a tidal power house equipped with tubular turbines can be provided with flatter foundations than the case of vertical shaft turbines. The wall of the power house of the basin side must be reinforced to cope up with the highest water level. Access to generators each enclosed in a bulb shaped casing is ensured by a service passage running along the whole length of the power house and housing ventilation ducts and electric cables.

DAM CONSTRUCTION

The dam may be either rock-fill or earths fill. The grading of fill material must be adopted to the increasing velocity of flow due to reduction of water passage during construction.

Another solution would be to sink prefabricated structural parts which would be floated to site on rafts or boats since a sea dam requires a much stronger construction. These hollow prefabricated structural part are sunk and assembled into two rows of piers braced by struts. The bottom blocks are fitted at their outside toes with hinged strips or grating which upon the block being in position are folded back to the sealed. The purpose of these is to minimise the effect of piping on the toe of the dam (Fig. 7).

Silting: The incoming tide waves bring lot of silt and detritus which settle down in the basin. This has to be removed periodically by dredging.

Navigation: The power installations set up at the mouths of the river, especially the dams of the basins close the waterway to navigation. Sea-locks have to be built for this purpose. It goes without saying that when determining the mode of operation of a tidal plant, not only must the technical aspects of energy potential, but also navigation requirements have to be considered.

THE RANCE TIDAL SCHEME

A tidal barrage across the mouths of the river Rance where it enters the sea is erected at St. Malo. The dam is 710 m. long, 48 m. wide and height of crest above the highest tide is 1.5 m. The generating unit takes the form of the totally submerged upstream bulb type (Fig. 8), there being 38 generating units immersed in orifices of the dam. Double carriage way at the top of the dam provides a bridge across the river. The dam contains a number of sluices, while on West bank, there will be locks for the use of vessels navigating the river (Fig. 9).

Turbine pumps are used to solve the problem of providing tidal power at all times of the day. The project is operated as two-way operating system.

The 38 generating units, each 9000 kW. have a total capacity of 342 M.W. and the annual output of 800 M. kWh in excess of the energy borrowed for pumping purposes. Cost of the project is £35 M.

The barrage will be made up as a hollow dam in R.C.C., in the interior of which are the 38 units. 10 large sluices (5 on each side) are arranged which help either to empty the reservoir or filling at high tide.

The mean tide height is 11.4 meters.

Another interesting tidal scheme proposed is the enclosure of "BAIE du SAINT MONT MICHEL", along Brittany coast, adjacent to St. Malo.

Dam lengths is 23.4 miles. It encloses a basin of 700 to 800 sq. km. at high tide level. 1200 generating units each of capacity between 7000 to 10,000 kVA are proposed. Total capacity is 8 million to 10 million kW. Annual production is 25 milliard (109) kWh. The difference in level between high tide and low tide reaches 13.50 m. (44 ft.) at exceptional tidal periods.

CONCLUSION

In view of the evergrowing energy requirements, the urgency of a thorough investigation of the potentialities of practicable tidal power schemes is hardly to be questioned. The advances in civil, electrical and mechanical engineering have already made such useful contribution that the first tidal plants are expected to function in the very near future. The problem of output regulation in tidal plants is satisfactorily met with by the adoption of turbine pumps. There is hardly any difficulty of civil engineering nature which cannot be overcome and hence we can conclude that both from the engineering and economic point of view, tidal power plants are today a perfectly practicable proposition to meet the growing demand for power. The experience gained in run-of-river plants can be practically applied in full to tidal schemes and in the support of theoretical and practical research, the dream of tidal plants can be brought to fruition.

It may be added that, the two dominating factors which intervene in tidal power schemes are the tide range and the area of power basin. It is therefore obvious that suitable sites are ever so much scarcer than is the case for normal river developments.

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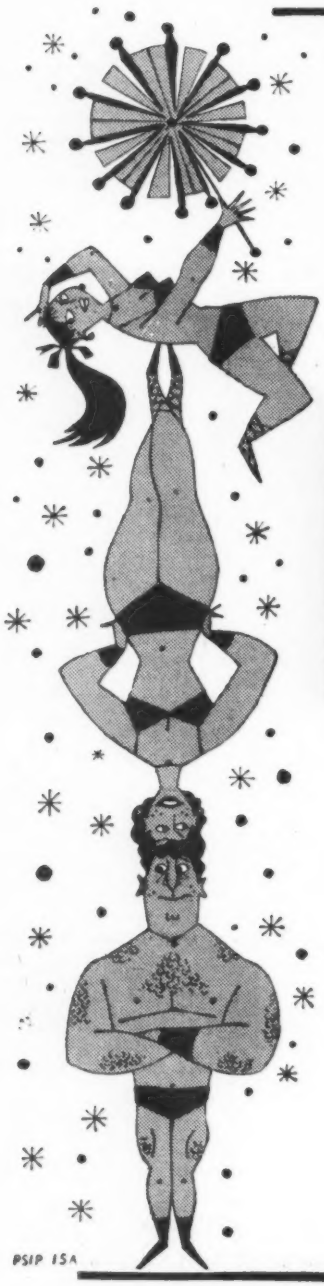

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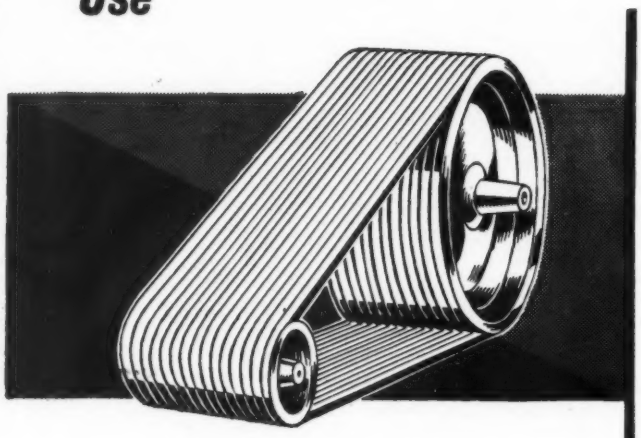

GOODYEAR
RAYON
V-BELTS

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GOODYEAR

THE GREATEST NAME IN RUBBER

A New Theory for Designing Axial Flow Pumps

By N. S. GOVINDA RAO

The success of the axial flow turbines gave an impetus to the design of axial flow pumps. Early designs were mostly empirical. Later (1928) pumps were designed on the one-dimensional theory [1, 2] This theory correlates with specific speed, the other characteristic non-dimensional ratios of the pump. It assumes one particular velocity to be characteristic of the entire flow through the hydraulic passges. The correctness of this approach is proportional to the effectiveness of the guidance of the water by the the vanes. Too many vanes to bring about correct guidance result in heavy friction losses. If the vanes are too few, one velocity cannot represent the entire flow between the vanes. This difficulty forced the development of the two-dimensional theory.

TWO-DIMENSIONAL THEORY EMBODIED IN AIRFOIL THEORY

The first comprehensive article on this theory was given by W. Spannhake in 1934 [3]. Inspite of considerable theoretical work having been done since then and before [4, 5, 6, 7, 8] the desing data is still largely dependent on experimental work for correct determination of the airfoils used and determination of lattice effect coefficients. The theory connecting the airfoil data with the design requirements is constantly undergoing improvement. The original work of Spannhake was based mostly on general engineering principles. Later on O'Brien and Folsom [9] developed a theory which took into account the drag coefficients as well. Other workers may also be mentioned [3, 6, 7, 8, 10 to 20]. Still later Wislicenus [21] in his book "Fluid Mechanics of Turbo-Machinery" (1947) derived his theory based purely on recognised principles of Fluid Mechanics. The present status of research is the development of the three-dimensional theory (21, 22). This is still in the early stages of development.

AIRFOIL THEORY

The procedure for designing vanes by this theory is briefly as follows. The flow conditions, that is the discharge and the particular head at which it is required are the main data. From the discharge the area of through flow is determined by assuming a suitable velocity. The first choice between the ratio of casing diameter to hub diameter is made generally on previous experience. Later it is corrected to fit in this parameter with other design and performance features. The assumed velocity V_{y_o} is then corrected after finalising the design by the equation

$$v_y = v_{y_o}/C_i \qquad .. \qquad (1)$$

where C_i. the thickness reduction ratio, can be correctly determined by the equation

$$C_i = \frac{t-T}{t} \qquad .. \qquad (2)$$

where T is the thickness of the vane. The space between the casing and the hub is now divided into a convenient number of equally spaced, cylindrical sections, at which the vane elements can be designed separately and which when combined will give the complete vane. Knowing the speed of rotation of the shaft, the linear speeds of the impeller at the chosen radii can be calculated. The head generated at all the radii is assumed to be the same and equal to the design head. From the head requirements of the pump, the peripheral component V_{u2} is determined using the Euler's equation

$$H = \frac{U_2 V_{u2}}{g} \qquad .. \qquad (3)$$

For design purposes, V_{u1} is assumed zero. Thus, at each of the chosen radii, the triangle of velocities are drawn (Fig. 1). The velocity of through flow, V_{F1} is kept constant at all radii but the tangential component will vary with the radii as H is constant. At any one of the radii chosen, a triangle of velocities is drawn (Fig. 1). V_∞ and β_∞ are then calculated as follows:

$$V_\infty^2 = V_F^2 + \left(U - \frac{V_{u2}^2}{2}\right) \qquad .. \qquad (4)$$

$$\text{and } \tan \beta_\infty = V_{F1}/U - \left(\frac{V_{u2}}{2}\right) \qquad .. \qquad (5)$$

Then the relation between coefficient of lift C_L and circulation is made use of as per the following equation:

$$C_L = \frac{2\Gamma}{v_\infty l} = \frac{2 \triangle V_x}{V_\infty} \frac{t}{l} = \frac{2 V_{u2}\, t}{V_\infty\, l} \qquad (6)$$

A suitable value of C_L is assumed and the t/l ratio is computed. Usually C_L is assumed as equal to or less than unity as this gives both a minimum value of C_D/C_L and smooth flow over the airfoil. Since the peripheral speed of the runner is higher at the casing than at the hub, C_L at the casing is much lower than at hub, for uniform lift. For values of C_L chosen, the airfoil corresponding to it is selected from the NACA Technical Reports No. 460. The angle of attack is known from these reports, and to this is added the angle β_∞ to obtain the vane

inclination (Fig. 2). The other remaining data, the length of the airfoil is obtained from the t/l ratio. The number of vanes used varies from 2 to 6, the lower number being used for high specific speed pumps. The vane spacing is calculated from:

$$t = \frac{2\pi r}{Z} \quad .. \quad (7)$$

where r is the radius of the section under consideration and Z is the number of vanes. Using this value of t in equation(7), the length is determined from the t/l ratio obtained from equation (6). This completes the design calculations for one section which is repeated for other sections.

LIMITATIONS OF AIRFOIL THEORY

This theory assumes frictionless, irrotational flow to occur through the hydraulic passages. This is not strictly true. The values of C_L and C_D are obtained from wind tunnel test data. These values may not be the same as when they are treated in a water tunnel. Due to the higher viscosity of water, the airfoils may be expected to stall at smaller angles of attack. It should also not be forgotten that tests in a wind tunnel are carried out on airfoils with a constant section along the span. In the pump, the section of the airfoil changes for each of the cylindrical radii chosen. This airfoil section of the pump varies in chord length and in inclination along the radius instead of being constart. It is also assumed that the velocity of through flow is constant at all radii and a zero pre-rotation is assumed. This is true to the extent the straight guide vanes before the impellers can help to achieve the condition. The drag of the fluid along the hub and the casing is neglected, as also the leakage losses around the blade tip clearances. One has to rely on only experience to fix the number of vanes to be selected, the optimum hub-casing ratio and even the diameter or speed of the impeller. The theory is applicable to points near to or at the design point (i.e. point of maximum efficiency). At lesser discharges, the angle of attack increases and the airfoil happens to stall under these conditions, the airfoil theory will no longer afford satisfactory results. It is for this reason that the discharge characteristics cannot be fully predicted by the use of the airfoil theory. Lastly in this theory, the total head on the pump is obtained by the weighted average of the heads generated at different radii. This method has been found to give inaccurate results when compared with experimental data.

However, despite these assumptions and limitations, the airfoil theory has *not yet been replaced completely* by any other theory. Current research centres round improvement in the airfoil theory like determination of cascade coefficients, development of more efficient vane forms, elimination of cavitation, pre-rotation etc.

FORCED VORTEX THEORY

Forced vortex theory as formulated at the Indian Institute of Science is indicated below :

It is well-known that if free vortex flow is assumed, for zero pre-rotation $H = U_2 V_{u2}/g$ = a constant (8)

But $U_2 = wr$ .. (9)

$$\therefore \quad H = \frac{WrV_{u2}}{g} = \text{constant} \quad .. \quad (10)$$

$$\therefore \quad v_{u2r} = Hg/w = \text{constant} \quad .. \quad (11)$$

Applying Bernoulli's equation, for a streamline flow in a free vortex:

$$H = P/r + V_u^2/2g = \text{a constant say } K_1 \quad .. \quad (12)$$

But $V_u r$ = a constant K_2. Substituting this in equation (11) We get:

$$H = P/r + K_2^2/2g.r^2 = p/r + K_3/r^2 = K_1 ; \; K_3 = K_2^2/2g$$

$$\therefore \quad P/r = K_1 - K_3/r^2 \quad .. \quad (13)$$

This equation indicates that in a free vortex motion, the static pressure varies inversely as the square of the radius. This assumption of flow in a pump casing to be that of a free vortex is subjected to the following drawbacks as indicated by Stepanoff (10).

(1) The free vortex motion of fluids can exist only at one point on the head discharge curve. At partial capacities, this free vortex law does not hold good and at zero capacity it is a forced vortex flow.

(2) It can be shown that the angular velocity of water varies inversely as the square of the radius

$$W = V_{u2}/r, \text{ But } V_{u2}V = K ; \; \therefore \; W = K/r^2 \quad (14)$$

It is difficult to understand how such an angular velocity distribution can result from constant angular velocity of the rotating impeller.

It is quite possible to generate different types of vortices within the pump casing. A more rational approach to the problem is to produce these vortices so that a definite relationship can be established between the type of vortex generated by a particular shape of an impeller and the shape of the discharge characteristic. This will incidentally bring about a co-ordination between theoretical design and test performance .

Table I gives the number and characteristics of the vortices selected for study. The characteristics include the following information: vorticity, angular velocity distribution, tangential velocity distribution pressure distribution and type of vortex.

In the above table, the laws of pressure distribution contain two constants C and n. These two constants have to be determined if the complete pressure distribution and tangential velocity distribution are to be known. Thus for two unknowns two conditions have to be sought, whereby the values of the two constants can be calculated. The first of the conditions

A New Theory for Designing Axial Flow Pumps

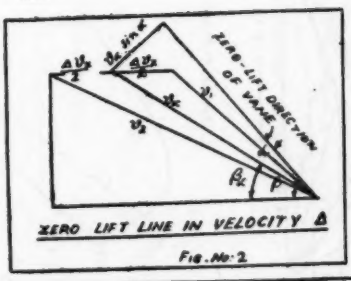


Fig. No: 2

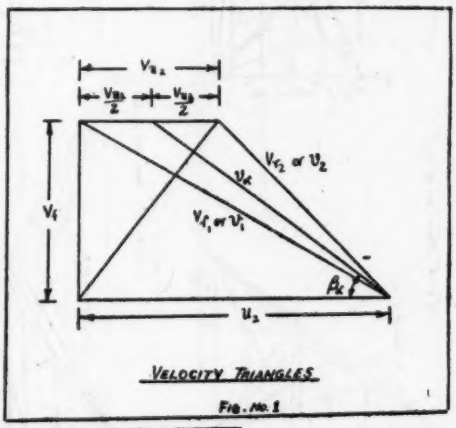
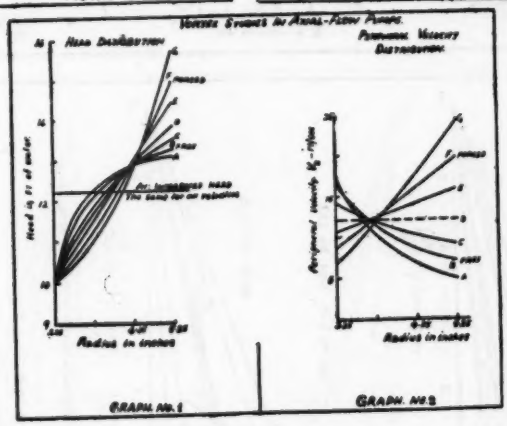


Fig. No. 1



VELOCITY TRIANGLES FOR IMPELLER 'B'

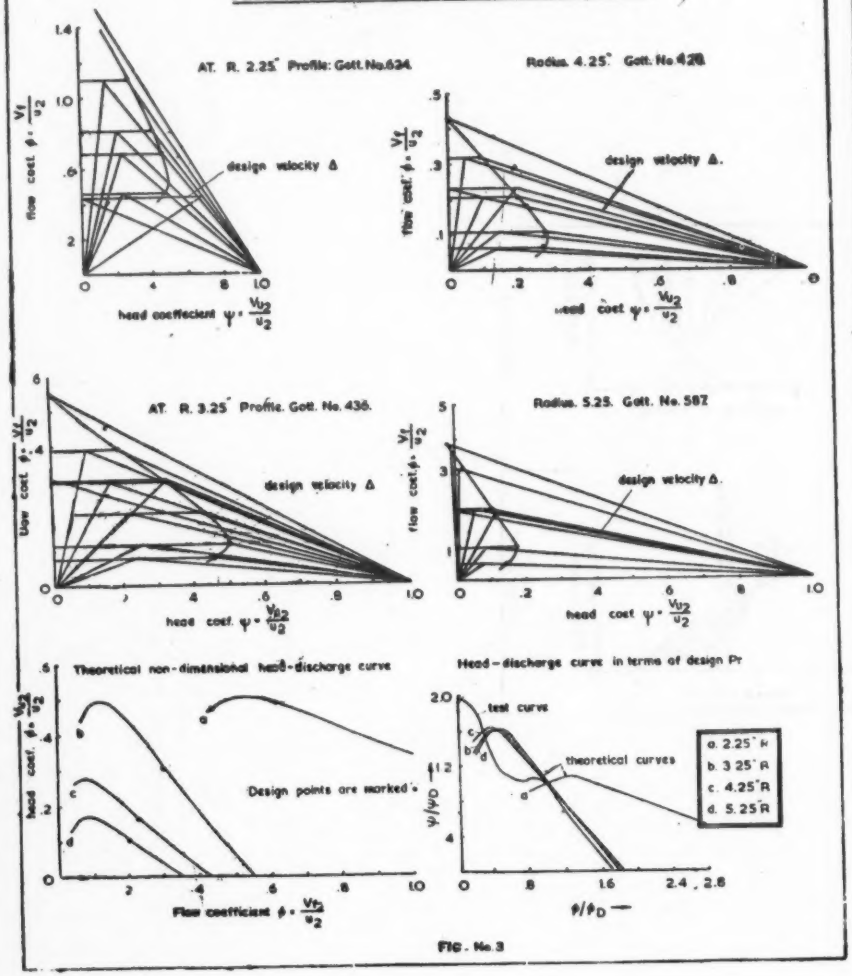


FIG. No. 3

VELOCITY TRIANGLES FOR IMPELLER C.

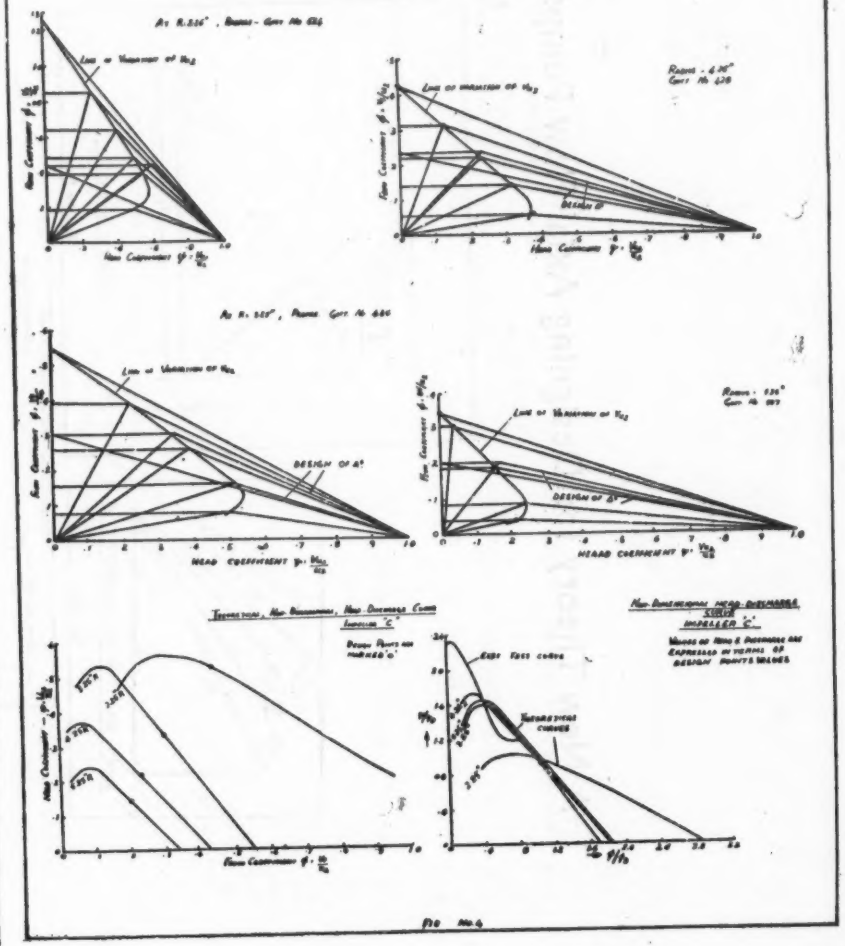


Fig No. 4

VELOCITY TRIANGLES FOR IMPELLER D.

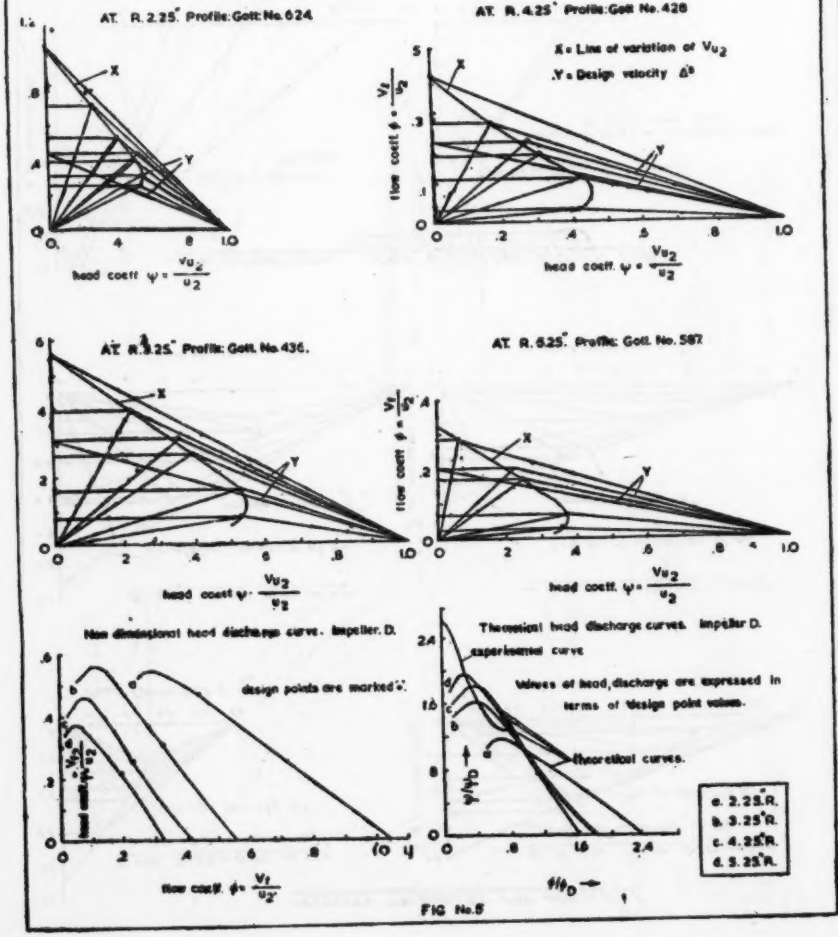

AT R. 2.25" Profile: Gott No. 624
AT R. 4.25" Profile: Gott No. 428
X = Line of variation of V_{u_2}
Y = Design velocity Δ^b
AT R. 3.25" Profile: Gott. No. 436.
AT R. 5.25" Profile: Gott. No. 587
head coeff $\psi = \frac{V_{u_2}}{u_2}$
flow coeff $\phi = \frac{V_f}{u_2}$
Non dimensional head discharge curve. Impeller D.
design points are marked a, b, c, d
Theoretical head discharge curves. Impeller D.
experimental curve
Values of head, discharge are expressed in terms of design point values.
theoretical curves.
ψ/ψ_D
ϕ/ϕ_D
a. 2.25°R.
b. 3.25°R.
c. 4.25°R.
d. 5.25°R.

FIG No.5

VELOCITY TRIANGLES FOR IMPELLER E.

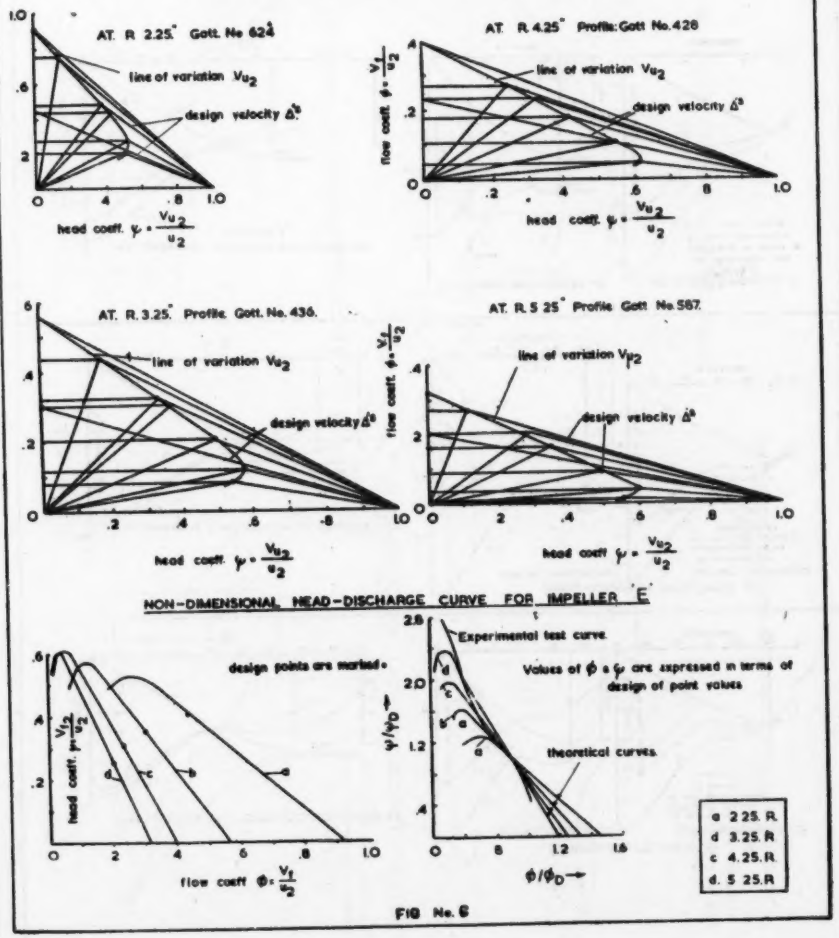
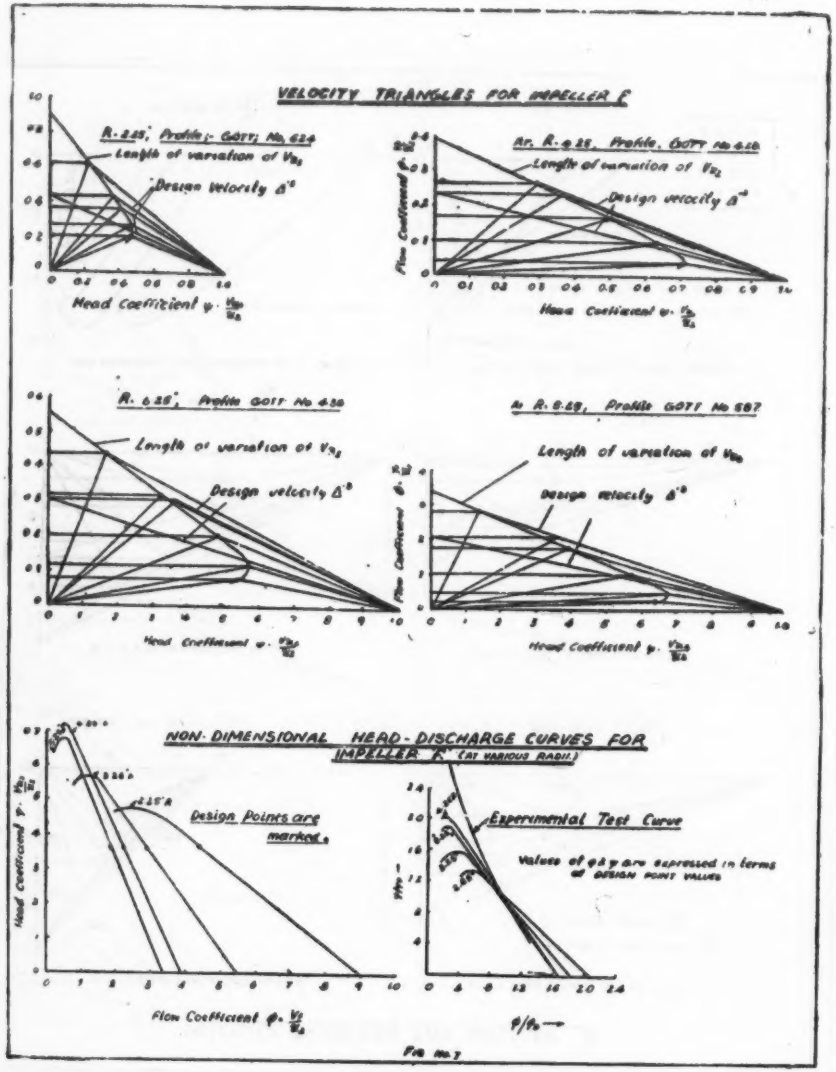
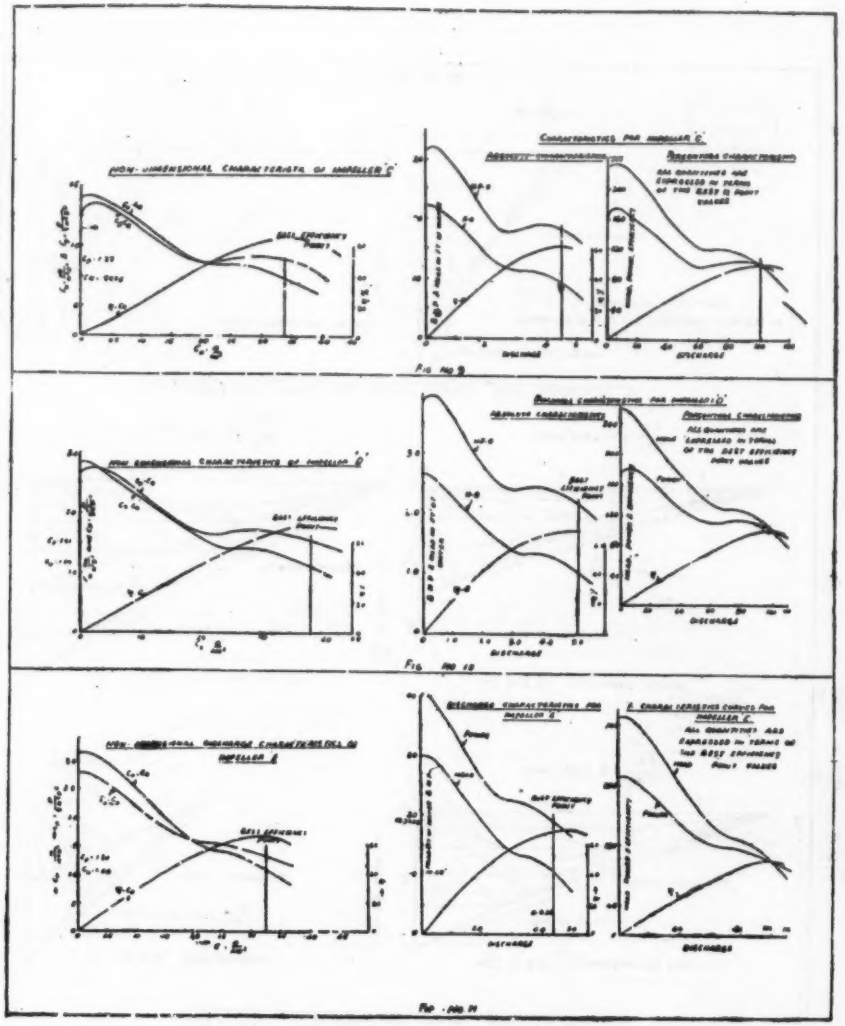

AT R 2.25" Gott. No 624
AT R 4.25" Profile: Gott No. 428
line of variation V_{u_2}
design velocity Δ^b
AT R. 3.25" Profile. Gott. No. 436.
AT R. 5.25" Profile Gott No. 587.
head coeff $\psi = \frac{V_{u_2}}{u_2}$
flow coeff $\phi = \frac{V_f}{u_2}$
NON-DIMENSIONAL HEAD-DISCHARGE CURVE FOR IMPELLER E
design points are marked a, b, c, d
Experimental test curve.
Values of ϕ & ψ are expressed in terms of design of point values
theoretical curves.
ψ/ψ_D
ϕ/ϕ_D
a 2.25. R.
d 3.25. R
c 4.25. R.
d. 5 25.R

FIG No. 6


VELOCITY TRIANGLES FOR IMPELLER E
R. 2.25", Profile :- GOTT; No. 624
Length of variation of V_{R_2}
Design Velocity Δ^0
Head Coefficient $\psi = \frac{V_{W_2}}{U_2}$
At R. 4.25", Profile GOTT No 620
Length of variation of V_{R_2}
Design velocity Δ^0
Flow Coefficient $\phi = \frac{V_{R_2}}{U_2}$
Head Coefficient $\psi = \frac{V_{W_2}}{U_2}$
R. 6.25", Profile GOTT No 626
Length of variation of V_{R_2}
Design velocity Δ^0
Head Coefficient $\psi = \frac{V_{W_2}}{U_2}$
At R. 8.25, Profile GOTT No 587
Length of variation of V_{R_2}
Design velocity Δ^0
Flow Coefficient $\phi = \frac{V_{R_2}}{U_2}$
Head Coefficient $\psi = \frac{V_{W_2}}{U_2}$
NON-DIMENSIONAL HEAD-DISCHARGE CURVES FOR IMPELLER 'E' (AT VARIOUS RADII)
Design Points are marked.
Experimental Test Curve
Values of ϕ, ψ are expressed in terms of DESIGN POINT VALUES
Head Coefficient $\psi = \frac{V_{W_2}}{U_2}$
Flow Coefficient $\phi = \frac{V_{R_2}}{U_2}$
$\psi/\psi_0 \rightarrow$
$\phi/\phi_0 \rightarrow$
FIG. NO 3


NON-DIMENSIONAL CHARACTERISTICS OF IMPELLER 'E'
CHARACTERISTICS FOR IMPELLER 'E'
ABSOLUTE CHARACTERISTICS
RELATIVE CHARACTERISTICS
ALL QUANTITIES ARE EXPRESSED IN TERMS OF THE BEST EFFICIENCY POINT VALUES
BEST EFFICIENCY POINT
DISCHARGE
NON-DIMENSIONAL CHARACTERISTICS OF IMPELLER 'F'
CHARACTERISTICS FOR IMPELLER 'F'
ABSOLUTE CHARACTERISTICS
RELATIVE CHARACTERISTICS
ALL QUANTITIES ARE EXPRESSED IN TERMS OF THE BEST EFFICIENCY POINT VALUES
BEST EFFICIENCY POINT
DISCHARGE
NON-DIMENSIONAL DISCHARGE CHARACTERISTICS OF IMPELLER E
DISCHARGE CHARACTERISTICS FOR IMPELLER 'E'
E. CHARACTERISTICS CURVES FOR IMPELLER 'E'
ALL QUANTITIES ARE EXPRESSED IN TERMS OF THE BEST EFFICIENCY POINT VALUES
BEST EFFICIENCY POINT
DISCHARGE
FIG. NO 11

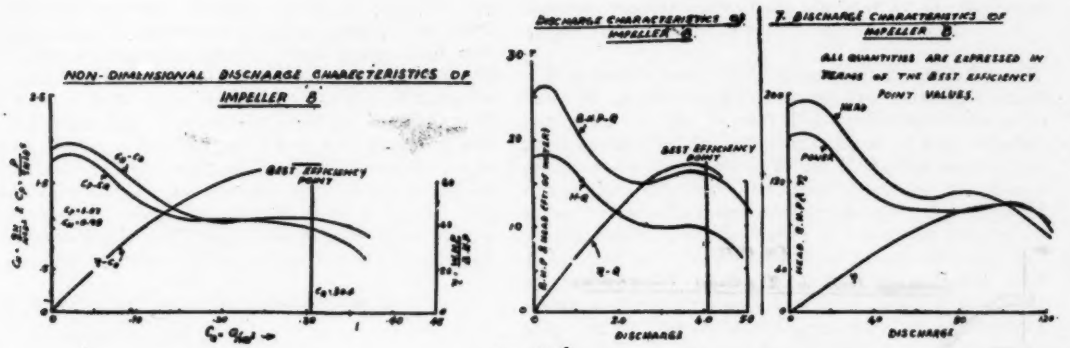


FIG. NO. 8

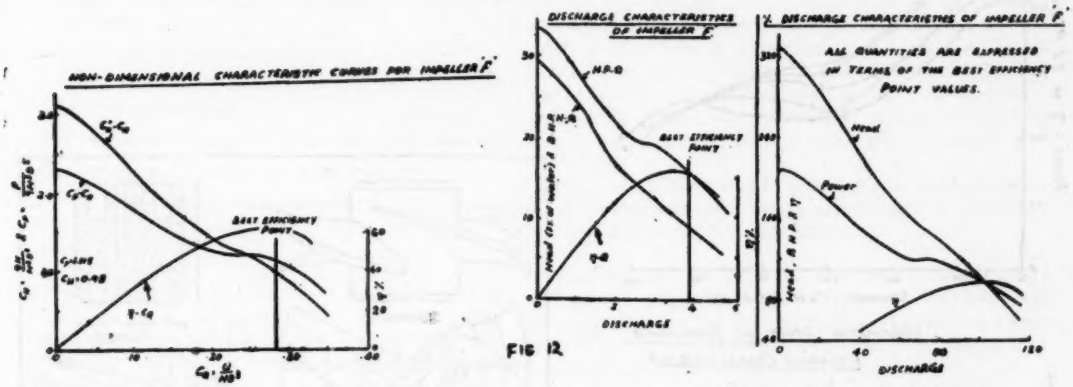


FIG 12

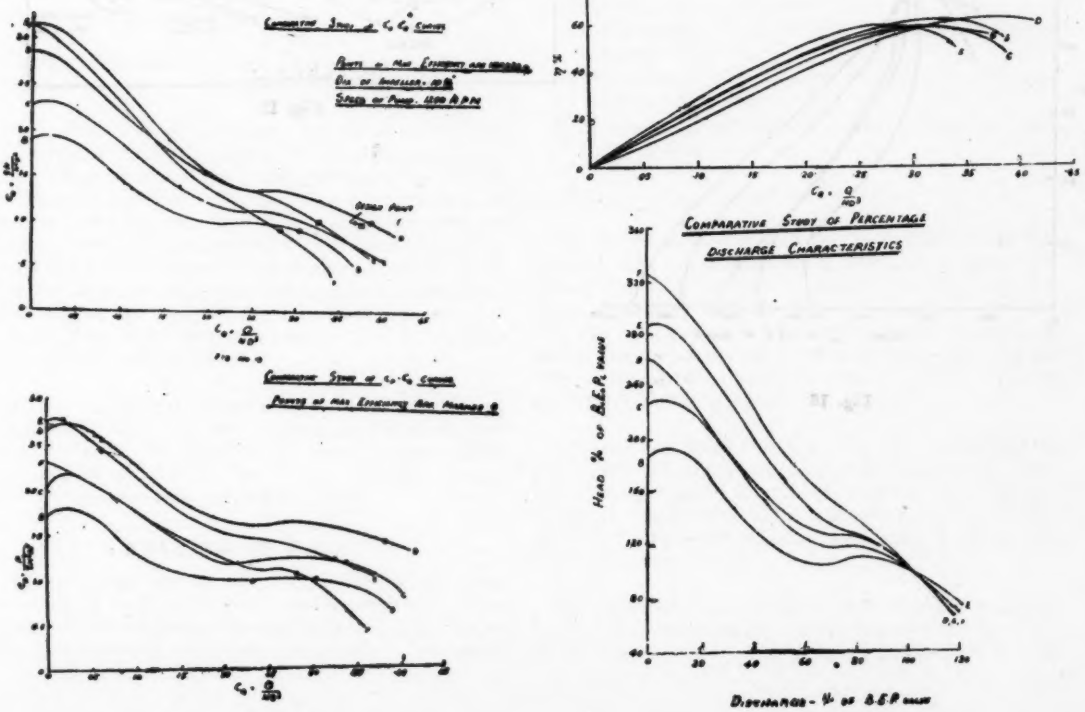

COMPARATIVE STUDY OF η-Cq CURVES
COMPARATIVE STUDY OF Cq-Cq CURVES
COMPARATIVE STUDY OF PERCENTAGE DISCHARGE CHARACTERISTICS
DISCHARGE - % OF B.E.P. VALUE

Fig. 14

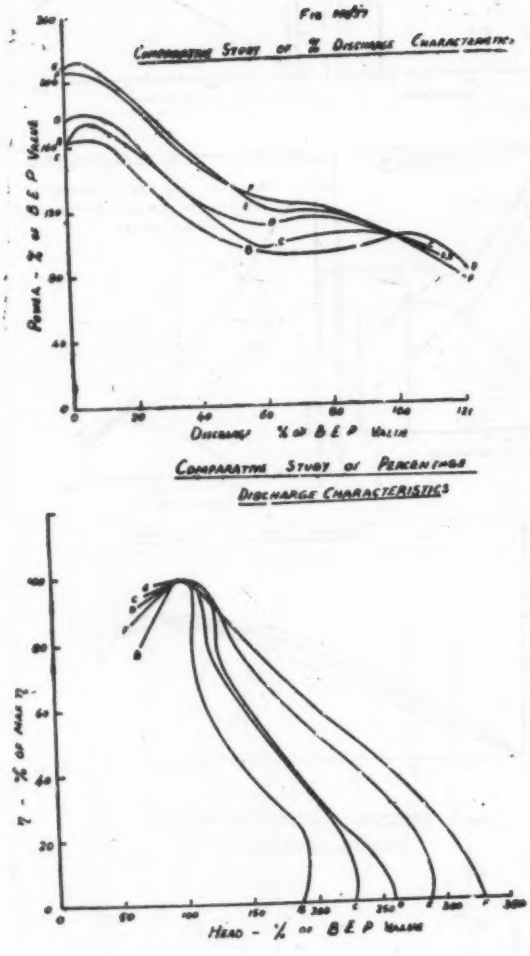

COMPARATIVE STUDY OF % DISCHARGE CHARACTERISTICS
POWER - % OF B.E.P VALUE
DISCHARGE % OF B.E.P VALUE
COMPARATIVE STUDY OF PERCENTAGE
DISCHARGE CHARACTERISTICS
η - % OF MAX η
HEAD - % OF B.E.P VALUE

Fig. 18

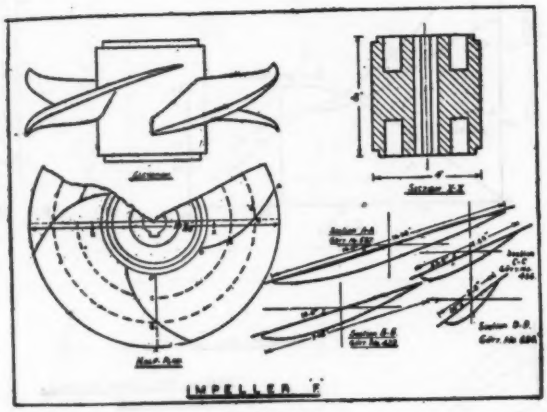

IMPELLER F

Fig. 19

is that the average, integrated H_m is equal to the design head. In the investigations made at the institute, the average integrated head h was kept constant at 12.5 ft. Secondly the head generated at the hub was made equal to 10 ft. for all types of vortices. Table 2 gives the characteristics of the vortices with values of C and h. The variations of pressure and tangential velocity with radius have been shown in Graphs 1 and 2. Seven designs were made based on the tangential velocity distribution given by the vortices in Table 2. Except this, all the other following parameters were held constant for all the designs.

(*a*) Number of design sections in the vane

(*b*) Average integrated head

(*c*) Velocity of through flow and hence the total discharge

(*d*) Speed of rotation of the shaft

(*e*) The four Gothingen airfoils selected for the four design sections

(*f*) The values of C_L at the four design sections

(*g*) The number of blades in each impeller

(*h*) In the plan view of all the impellers, the leading edges of the airfoil were kept along one straight radial line.

By keeping these factors constant, the radial rate of variation of head in the pump was allowed to manifest itself in two major design parameters. The two parameters are the length of the airfoil and the inclination of the airfoil at the design section. These parameters influence the final complete vane inasmuch as the length determines the vane shape and the variation in the inclination determines the twist in the vane from the hub to the casing.

Complete designs for seven impellers were worked. Table 3 gives abstracts of design of impeller 'F'.

The pump in which the design impellers were tested was specially manufactured for this purpose. The general dimensions of the pump necessary for testing the impeller designs were as follows:—suction pipe diameter $10\frac{1}{2}$ in. Impeller casing diameter $10\frac{1}{2}$ in. Hub diameter $4\frac{1}{2}$ in. Delivery pipe diameter 12 in. Speed of pump 1200 rpm. The design head was 10 ft. of water and the discharge 1870 gpm. Specific speed 9270. Water HP 5.67 HP and the average velocity of through flow 10.2 ft/sec. Assuming $C_i=0.9$, we have $VF=10.20.9=11.35$ ft/sec.

All the vanes were designed at four cylindrical sections, one at the hub, one at the casing, and at two other sections. The radii of the sections were $2\frac{1}{4}$ in, $3\frac{1}{4}$ in, $4\frac{1}{4}$ in and $5\frac{1}{4}$ in. The design of guide vanes does not call for any special comment.

ANALYSIS OF DATA

Actual pumps were constructed with these designed impellers A to F and their characteristics curves drawn. The actual experimental values obtained have been superposed on theoretical values obtained from purely theoretical calculations for the different impellers in Figs. (3) to (7).

It is easy to see that in the case of impellers E and F, the test curve is quite different from the theoretical curves. Thus, the Airfoil Theory formula cannot be used to predict the characteristics of any impeller designed on the vortices E and F. In the case of the remaining impellers, the prediction of the head-capacity curve is accurate enough at the design point and beyond (i.e. for discharge values higher the design value). It is to be noticed that the average curve of the radii at 5.25", 4.25 and 3.25, is to be taken. The curve at R=2.25" does not help in creating pressure. The Airfoil Theory fails to predict the characteristics at smaller discharges because this theory does not hold good after the airfoils have stalled. It is to be remembered that the characteristics of the airfoils used were obtained from wind tunnel data and it is known that due to viscosity the same airfoils tested in water (i.e. hydrofoils) will stall earlier. Use of water-tunnel characteristics of airfoil will possibly yield better and more accurate results in prediction.

At present, the airfoil theory can be used to determine the characteristics of an impeller only in the range of high discharges and low heads.

The test-curves obtained for individual impellers are given in Figs (8) to (12). These results have been summarised in a few graphs in Figs. (13) to (18). These curves will be studied individually as important conclusions could be derived from them.

(*a*) C_H-C_Q curve (Fig. 13); (*b*) C_P-C_Q curve (Fig. 14);

(*c*) $\eta-C_Q$ curve (Fig. 15)

(*d*) Percentage of power—Percentage of discharge curve (Fig. 16).

(*e*) Percentage of head—Percentage of discharge curve (Fig. 18).

(*f*) Percentage of efficiency—Percentage of head curve (Fig. 18).

(*a*) Study of C_H-C_Q curve (Fig. 13):

The first conclusion that can be reached from this graph is that the inclination of the C_H-C_Q curve is a direct function of the vorticity of the impeller. Thus, impeller F with the largest amount of vorticity has the steepest C_H-C_Q curve, while impeller B which has zero vorticity has the flattest curve of all. The intermediate vortices have curves varying uniformly between these two limits.

The shut-off head (i.e. C_H at $C_Q=0$) is the highest for impeller E and is the lowest for impeller B. The low shut-off head of impeller B is of very great importance as it indicates that it will also have a low shut-off B.H.P. This will be confirmed when studying the C_P-C_Q curve.

The most efficient point of each impeller has been indicated on each graph. The position of the points is interesting to study. The points for impellers B, C, E

and F are close together while the best efficiency point for impeller D is isolated. This shows that vortex D is most suitable for designing axial pumps for very high discharges and very low heads. The other vortices give a lower discharge at the same head.

The position of these best efficiency points in relation to the design point gives a measure of the designing ability of an engineer. All the impellers have been designed for the point, at $C_Q = .373$ and $C_H = 1.05$. In the case of impeller D, the best efficiency point head and discharge are greater than the design values. Hence, a designer would do well to assume a smaller manometric efficiency for designing purposes. The best efficiency points for impellers B and F are most distant from the design point.

(*b*) Study of C_P—C_Q curves (Fig. 14):

The impeller with the highest shut-off B.H.P is impeller E and impeller B gives the least shut-off B.H.P. Also, impeller B gives the flattest C_P—C_Q curve and hence it is the impeller which is to be recommended when there is a valve in the delivery circuit. Although the curve for impeller F has a small shut-off head, it is very steep and at high values of C_Q delivers water with the least B.H.P. input.

The best efficiency points for impellers B and F occur at small values of C_Q and C_P while the best efficiency point for impeller D occurs at relatively high values of C_Q and C_P.

(*c*) Study of, η—C_Q curves (Fig. 15):

At small values of C_Q the impeller F gives the best efficiencies, as it is flatter than the others. However, impeller F does not have a high maximum efficiency and the maximum efficiency occurs at a wall value of C_Q.

Impeller D possesses the maximum efficiency and it occurs at a high value of C_Q. Impeller B appears to have the lowest efficiency but is flatter than all the other in the region of the maximum efficiency.

(*d*) Study of percentage of head—Percentage of Q curves (Fig. 16):

The percentage characteristics are very useful to study as their geometry is independent of the design point values. That is to say, even if another axial-flow pump were designed for a higher or lower discharge, on any one vortex, the geometry of its percentage of head—percentage of Q characteristic would be identical with the shape indicated for that vortex as shown.

The curves become more and more steep as the vorticity of the impellers increases. Thus, impeller F has the steepest curve and impeller B B the flattest. The shut-off head is as high as 326 per cent of the design head for impeller F, as against 188 per cent for impeller B. The shut-off head for the other impellers increases uniformly. In the case of impeller B, the curve is very wavy, i.e. as the valve is opened from its closed position, the percentage head first increases, then sharply falls, increases once again before the design point and drops again after the design point.

(*e*) Study of percentage of B.H.P—Percentage of discharge curve (Fig. 17)

This is one of the most important graphs, as it is from these curves that the size of the motor for the pump is to be decided. The curves indicate that the impeller E requires a motor whose H.P. has to be 212 percent of the design power of the pump. This figure is brought down to 165 percent in the case of impeller B. It is to be noticed that the maximum percentage of B.H.P occurs at a point a little before complete closure of valve. Once again, impeller B has the flattest curve, for even at 40 per cent of design discharge, the B.H.P. is 108 per cent the design B.H.P.

Thus in all situations where a valve has to be operated in the delivery circuit, impeller B is to be recommended as it has a flat B.H.P—Q curve.

(*f*) Study of percentage of efficiency—Percentage of head curve (Fig. 18)

These curves have been plotted to bring out the advantage of impellers E and F. Many situations occur in practice where the pump has to deliver water over a wide range of heads. In these circumstances the disadvantage of using impeller B could be very easily seen from this figure. For small ranges of head about the design point, impeller E is seen to have the flattest η—H curves. For very wide ranges of head, impeller F is to be favoured.

Conclusion

Axial-flow pumps have to be designed according to the specific use to which they will be put. Some have to deliver very large quantities of water at very low heads. Others have to deliver a large volume of water at a very low heads. Others have to deliver a large volume of water at a high head. Still others have to operate with high efficiency over a larger range of heads. Each of these conditions have to be met with a special design of impeller. The answer to this requirement is provided by the use of the vortex theory of axial-flow pumps.

The airfoil theory as it exists to-day, does not allow the designer much scope for producing various shapes of impellers. This is because the head generated along the radius is kept constant. The vortex theory permits the head to be varied according to the formulae provided by each vortex. Thus, the vortex theory enables one to design pumps with a varying amount of vorticity in the pump.

In this way, it has been possible to design different impellers for the same head and capacity requirements. The characteristics of these impellers have indicated many striking features.

Impeller B, the free-vortex impeller, has the outstanding advantage of having a flat power vs discharge curve and the lowest value of power for nil discharge.

Impeller D, the impeller which generates a constant peripheral velocity along the radius, could be used most advantageously when the pump requirements call for large quantities and high pressures. This impeller has the highest efficiency compared with the others.

Impeller F, the forced-vortex impeller, has the advantage of operating with higher efficiency at partial discharges. It is particularly to be chosen when the pump has to work against a wide range of head.

Thus, it is seen that the vortex theory of axial-flow pumps enables the engineer to design axial flow pumps for different requirements. It is the versatility of this designing procedure that will make the vortex theory more useful than the airfoil theory. The efficiencies given in the table can be considerably improved by better workmanship from what was possible in a small workshop.

Acknowledgements

Much of the credit for developing this theory should go to Sri D. N. Contractor. The author wishes also to acknowledge the help of Sri K. Seetharamiah at all stages both during the development of the theory and the experimental work thereon.

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Table 1

Sl. No.	*Impeller Number*	*Vorticity z*	*Angular velocity distribution*	*Tangential velocity distribution*	*Pressure distribution*	*Type of vortex*
1	A	$_z=\frac{C_1 r^{-5/2}}{2}$	$w=C_1 r^{-5/2}$	$V_u=C_1 r^{-3/2}$	$\frac{P}{\gamma}=\frac{-C_1^2}{3gr^3}+h_1$	
2	B	$_z=0$	$w=C_2 r^{-2}$	$V_u=C_2 r^{-1}$	$\frac{P}{\gamma}=\frac{-C_2^2}{2gr^2}+h^2$	Free vortex
3	C	$_z=\frac{C_3 r^{-3/2}}{2}$	$w=C_3 r^{-3/2}$	$V_u=C_3 r^{-1/2}$	$\frac{P}{\gamma}=\frac{-C_3^2}{gr}+h_3$	
4	D	$_z=C_4 r^{-1}$	$w=C_4 r^{-1}$	$V_u=C_4 r^{\circ}=C_4$	$\frac{P}{\gamma}=\frac{C_4}{g}\log_e r+h_4$	V_u is constant
5	E	$_z=3/2C_5 r^{-1/2}$	$w=C_5 r^{-1/2}$	$V_u=C_5 r^{1/2}$	$\frac{P}{\gamma}=\frac{C_5^2 r}{r}+h_5$ V_u^2/r	Centrifugal force=constant
6	F	$_z=2C_6$	$w=C_6 r^{\circ}=C_6$	$V_u=C_6.r$	$\frac{P}{\gamma}=\frac{C_6^2 r^2}{2g}+h_6$	Forced vortex
7	G	$_z=5/2C_7 r^{1/2}$	$w=C_7 r^{3/2}$	$V_u=C_7 r^{3/2}$	$\frac{P}{\gamma}=\frac{C_7^2 r^3}{3g}+h_7$	Superforced vortex

Table 2

Sl. No.	*Vortex Ref.*	*Vorticity*	*Law of angular velocity distribution*	*Law of peripheral velocity distribution*	*Law of pressure distribution*	*Type of vortex*
1	A	$_z=-.715\ r^{-5/2}$	$w_z=1.43\ r^{-5/2}$	$V_u=1.43\ r^{-3/2}$	$\frac{P}{\gamma}=\frac{-.02222}{r^3}+13.36$	
2	B	$_z=0$	$w_z=3.02\ r^{-2}$	$V_u=3.02\ r^{-1}$	$\frac{P}{\gamma}=\frac{-.1418}{r^3}+14.035\ h+\frac{P}{\gamma}+\frac{V^2}{2g}$	Constant $\therefore$ Free vortex
3	C	$_z=3.065\ r^{-3/2}$	$w_z=6.13\ r^{-3/2}$	$V_u=6.13\ r^{-1/2}$	$\frac{P}{\gamma}=\frac{-1.168}{r}+16.235$	
4	D	$_z=12.2\ r^{-1}$	$w_z=12.2\ r^{-1}$	$V_u=12.2\ r^{\circ}=12.2$	$\frac{P}{\gamma}=4.63\log_e r+17.74$	V_u is constant along radius
5	E	$_z=36.0\ r^{-1/2}$	$w_z=24.0\ r^{-1/2}$	$V_u=24.0\ r^{1/2}$	$\frac{P}{\gamma}=17.93\ r+6.58$	V_u^2/r=centrifugal force=constant
6	F	$_z=90.8\ r^{\circ}=90.8$	$w_z=45.4\ r^{\circ}=45.4$	$V_u=45.4\ r$	$\frac{P}{\gamma}=32\ r^2+8.875$	w_z=constant $\therefore$ forced vortex
7	G	$_z=211.25\ r^{1/2}$	$w_z=84.5\ r^{1/2}$	$V_u=84.5\ r^{-1/2}$	$\frac{P}{\gamma}=74.3\ r^3+9.50$	Superforced vortex

Table 3

Summary of Design of Impeller 'F'

Sl. No.	*Quantity*	*Sections*			
		1	2	3	4
1	Radius	2.25″	3.25	4.25″	5.25
2	u_2	23.6	34.05	44.5	55.0
3	Head	10.00	11.225	12.885	15.0
4	V_{u2}	8.51	12.3	16.1	19.86
5	V_∞	21.57	30.1	38.25	46.5
6	β_∞	31.75	22.15	17.28	14.14
7	C_L 1/t	.79	.818	.843	.854
8	C_L	1.0	0.9	0.8	0.7
9	l/t	1.267	1.1	0.95	0.82
10	K	1.06	1.43	2.00	1.82
11	α	3.3	1.428°	.65	2.5°
12	$\beta = \beta_\infty - + \infty$	35.05	23.58°	16.628°	16.62°
13	t_4	3.535″	5.11″	6.68″	8.25″
14	l_4	2.79″	4.65″	7.04″	10.06″
15	Depth of elevation 1Sinβ	1.6″	1.85″	2.01″	2.875″
16	Plan length. 1 Cosβ	2.285″	4.26″	6.74″	9.63″
17	Airfoil Gottingen	624	436	428	587

Table 4

Test results of Impeller 'F'

Sl. No.	*Speed*	*Head*		*Discharre*			*Power*									
		Ins. of Hg.	*Ft. of H_2 0*	*Point gauge reading*	*Head over wire*	*Q from graph*	*Stage of rheostat*	*Amps*	*K.W.*	*Eff. of Motor*	*BHP K.W.x/.746*	*WHP WQH/550*	*Eff. WHQ/BHP*	$C_Q = Q/ND^3 = Q/13.4$	$C_H = \frac{gH}{H^2D^2} = H/9.5$	$C_P = \frac{P}{\rho N^3 D^5} = P/14.44$
1	1225	5.40	5.67	1.85	15.24	4.6	13	7.6	4.32	86.0	4.96	2.96	59.5	.336	.573	.881
2	1235	6.50	6.82	2.14	14.95	4.39	13	8.64	4.93	86.0	5.68	3.395	59.77	.3185	.715	.878
3	1180	7.20	7.56	2.64	14.45	4.03	13	8.26	4.71	86.0	4.53	3.344	63.62	.396	.823	1.006
4	1170	7.65	8.03	2.94	14.15	3.82	13	8.38	4.87	86.0	5.61	3.48	62.0	.2925	.889	1.074
5	1190	9.40	9.87	3.33	13.76	3.57	14	9.12	5.75	85.5	6.58	4.09	69.79	.269	1.058	1.168
6	1255	11.90	12.50	3.64	13.45	3.36	15	11.50	7.11	85.5	8.148	4.76	58.42	.24	1.204	1.18
7	1200	13.70	14.40	5.27	11.82	2.48	15	11.20	7.00	85.5	8.02	4.05	50.50	.1852	1.518	1.388
8	1190	14.05	14.75	5.50	11.59	2.35	15	11.10	6.94	85.5	7.95	3.93	49.43	.177	1.578	1.42
9	1170	15.50	16.28	6.33	10.76	1.945	15	11.21	7.02	85.5	8.03	3.59	44.70	.145	1.80	1.54
10	1180	14.85	15.60	6.10	10.99	2.05	15	11.20	7.00	85.5	8.02	3.625	45.19	.1555	1.695	1.484
11	1170	16.90	17.73	6.99	10.10	1.675	15	11.70	7.33	85.5	8.40	2.97	35.35	.1282	1.962	1.603
12	1215	18.40	19.32	6.82	10.27	1.74	16	8.39	8.39	84.0	9.38	3.81	40.31	.1282	1.995	1.56
13	1180	19.90	20.9	7.76	9.33	1.38	16	-	8.33	84.0	9.38	3.265	34.80	.1048	2.27	1.735

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Centrifugal and Propeller Pumps: Operating Characteristics and Preventive Maintenance

By LEWIS HANFORD KESSLER

It is indeed a privilege to have been asked to prepare a paper for the Civil and Hydraulic Engineering Section Seminar on Hydraulic Manchines for your Golden Jubilee Celebration. I regret that I cannot appear in person to enjoy fully the occasion and to meet personally you people, some of whom are known to me only by your handiwork.

It occurred to me that probably a number of waterworks or utility men would be present and that I might discuss some hydraulics that could assist them in knowing more about pumping machinery. Then I could include some of the "why it was designed the way it is" and some reasons for the selection of pumps for a given task. In America, I have found an increasing interest in some workable preventive maintenance program for mechanical equipment. Because of my rather broad experience in this matter during World War II, I am offering some suggestons as to a program that will obtain maxmum service of machinery at minimum expense.

Researches indicate that in 1754 Euler in Germany developed a centrifugal pump in primitive form. It was never put into practical use. During the next 100 years, a number of rotary and centrifugal pumps were invented largely by French engineers. Good performance was not obtained. A Mr. MacCarthy in the year 1830 made a pump and erected it in the New York Navy Yard. At the middle of the 19th century (1851) J. G. Appold of London, England, achieved great success with his centrifugal pump because he obtained efficiencies almost three times the magnitude ever reached by former experimenters, or a jump from about 25% to 65%.

Obviously, the pump is not a modern invention, but since the beginning of the 20th century, great strides have been made in attaining outstanding operation and performance. In some cases maximum performance may have been obtained. This means that the designer has embodied within his pump all of the established theory on flow and transformation of energy and that he has reduced mechanical and hydraulic losses to a minimum. Plate I graphically portrays a summary of accomplishment.

It is manifest that as the centrifugal pump developed, it had to compete with the already established reciprocating pumps of high efficiency. This competition has been successful because the centrifugal pump operates at relatively high speeds and is usually (except in case of the largest propeller types) direct connected to the driver or prime mover so that power transmission losses are small. Because of their high speed, the the pumps are relatively small as compared with reciprocating machines. Some of the advantages of the centrifugal pump which have given it wide application are:

1. Low initial cost
2. Low installation cost
3. Low housing cost
4. Low operating cost
5. Low maintenance cost
6. Simple regulation
7. Direct connection to driver or prime mover, usually eliminating speed reducers.
8. Pumping any liquid including sewage.

PUMP EFFICIENCY

The efficiency of a centrifugal pump depends upon the reduction of the following inherent losses:

1. Hydraulic
   - *a*. Friction loss in impeller.
   - *b*. Friction loss in volute and/or diffusion vanes.
   - *c*. Shock losses in volute and/or diffusion vanes.
   - *d*. Wearing ring leakage.
   - *e*. Disc Friction Losses.
2. Mechanical
   - *a*. Bearing friction losses
   - *b*. Stuffiing box losses.

Relative to performance, the efficiency of a pump depends upon, CAPACITY (discharge), HEAD pumped against, and SPEED. Of these three factors, hydraulic capacity is the most important. Friction and turbulence losses are a smaller percentage of the total required power when larger capacities are encountered. Disc friction and mechanical losses are relatively smaller at large capacities. It is to be expected that increased head for a given capacity will result in lower efficiency as increased head requires either a larger diameter impeller or a higher speed. In either case, increased mechanical losses result.

PUMP CLASSIFICATION

Indications are that the present application of centrifugal pumps is so broad that it can be said it has very few, if any, limitations. This statement is made as a result of the many variations in type and classification as illustrated in the Figure 1 to 6. They are described in brief as follows:

Figure 1—Radial Type Impeller. Head is developed largely by action of centrifugal force and is used for medium and high head work at relatively low capacities. It is truly the conventional type and appears in almost all multi-stage designs. Ratio of the discharge diameter to the inlet diameter is about 2.0 or slightly greater.

Figure 2—Double Suction Type Impeller. Larger capacities are obtained at about the same head range as for the single suction impeller of Figure 1. This type is hydraulically balanced as the end thrusts are opposed and in balance. This added advantage for certain purposes should not be overlooked.

Figure 3—Curved Blade or Francis Type Impeller. It is used either for lower heads or higher capacities. Ratio of discharge diameter to the inlet diameter is usually much smaller than for Figure 1 and Figure 2. For a given capacity and head the Francis impeller can operate at higher speed than the conventional radial blade impeller (Figure 1). Impeller can be constructed as a double suction also.

Figure 4—Mixed Flow Francis. Head developed by this type is caused in part by the centrifugal force and the remainder is due to the push of the blades. Discharge is partly radial as well as axial and hence the reason for the name of mixed-flow. Mean discharge diameter is about equal to the inlet diameter.

Figure 5—Propeller Type. Almost all of the head developed is caused by the push of the blades as the flow is almost entirely axial. Heads are low with a range of 3 to 30 feet, speed is from 100 to 1800 rpm and capacities are exceptionally large up to 800 cfs or greater.

SPECIFIC SPEED, N_s

To study any or all types of centrifugal pumps, whereby a small pump becomes a model for a larger pump, or the larger one is homologous to the smaller, a most useful type coefficient must be introduced in this place, the Specific Speed. It is definitely a type index or a numerical classification (not dimensionless) of impellers based entirely on their test performance and geometrical proportions regardless of their actual size or the speed at which they operate.

Definition: Specific speed is the speed in rpm at which an impeller would operate, if reduced proportionately in size so that it will deliver a rated capacity of 1.0 GPM against a total Head of 1.0 feet. It is designated by symbol N_s and is determined by the formula

$$N_s = n \frac{\sqrt{Q}}{H^{3/4}}$$

Where n = revolutions per minute

Q = capacity or discharge in gallons per minute

H = Head in feet

$H^{3/4}$ = is found by $\frac{H}{\sqrt{\sqrt{H}}}$

OR $\left\{ \sqrt{\sqrt{H}} \right\}^3$ Require only one motion of slide rule cursor (glass)

Attention is invited now to Figures 1 through 6 to note the specific speed range or type index indicated for each impeller type as well as plate 2.

A SECONDARY CLASSIFICATION

Centrifugal pumps can be classified according to specific speed or magnitude of pressure into three groups as follows:

	For Given Speed and Capacity	*For Given Speed and Pressure*	*For Given Capacity and Head*
1. Low N_s	High Pressure	Low Capacity	Low Speed
2. Medium N_s	Medium Pressure	Medium capacity	Medium Speed
3. High N_s	Low Pressure	High Capacity	High Speed

1. Low Pressure.....2′ to 60′ propeller and mixed flow.
2. Medium Pressure...60′ to 200′ radial and Francis type
3. High Pressure.....200′ and above, radial or partial Francis type.

THEORY

The centrifugal pump as the name implies depends largely on the action of centrifugal force imparted on the water by the rotating blades or vanes of the impeller for the development of the head pumped against. It is truly a "velocity" pump.

We know from physics that the velocity of a falling body (water in this case) rather than the solid, if not impeded by some resistance, is accelerated to a value of:

$V = g \times t$ where g = acceleration of gravity = 32.2 ft./sec.^{2}

$t = \frac{V}{g}$ t = time interval of motion in secs.

and the actual distance (S) or (h) through which the liquid falls with uniform acceleration caused by gravity is:

$$S = \tfrac{1}{2} g t^2 = h$$

Substituting $t = V/g$ in the above equation, we have

$$h = \tfrac{1}{2} g \left(\frac{V}{g} \right)^2 g = \frac{V^2}{2g} \text{ or } V = \sqrt{2gh} \quad (1)$$

Where V = absolute velocity in ft./sec.

h = head or total available pressure in feet of water.

It is manifest that velocity is a function of the head (or pressure) or in other words, the higher the head the higher the velocity.

From the above where $h = \frac{V^2}{2g}$ (Velocity head or kinetic energy in fts lbs. per lb. of liquid pumped)

This is identical with equation...(1) and shows that pressure head can be transformed into velocity head and volocity head can be transformed into pressure head. Losses in head are always involved.

We must recognize that the velocities in high pressure (low specific speed) pumps reach enormously high mangnitudes and this fact results in high friction losses. Unfortunately, in hydraulics it is always more difficult to transform velocity energy into pressure energy such as in a venturi, than it is pressure energy into velocity energy as in a nozzle.

While the above figures provide some concept as to what the hydraulic machine is trying to do, perhaps the comparison of two pumps of indentical capacity but with a requirement of difference in head will help you visualize velocities created within pumps of various conditions

Pump No. 1—45,000 GPM @215′ head—Average Velocity in volute=47 fps.
Pump No. 2—45,000 GPM @ 45′ head—Average Velocity in volute=22 fps.

SUCTION LIFT AND SUCTION HEAD

In the design of pumps the designer must give due consideration to the atmospheric pressure under which it must operate. A pump not only delivers its discharge against a pressure but it also may be required to draw its supply from a suction bay or well. The pump may also be required to handle water at much higher than ambient temperatures or at elevations much higher than sea level. In either case, the impeller inlet is made to secure the necessary capacity. We must keep in mind that there are two possible operating conditions for the intake.

1. Suction lift (vacuum)
2. Suction head (pressure)

Both cases have a direct effect upon the general performance of a pump and must be clearly analyzed before a design and installation is made. The suction lift especially must be kept within certain limits to prevent creation of vapor in the suction passage and eye of impeller and subsequent cavitation.

The specific speed of a pump and the head at which it must operate is an index to the allowable or safe suction condition (head or suction lift) at which it will operate without undue cavitation.

Cavitation:— An operating condition all designers and water works operators desire to eliminate is what has come to be known as cavitation. Caivitation occurs in flowing fluids, but in particular in flowing liquids. It consists of two separate and distinct phenomena, *First* the formation of vapor bubbles caused by a localized reduction in pressure below the boiling point of the liquid pumped and *Second* the collapse of the vapor bubbles as the pressure increases again. It is the second stage of the phenomena, the collapse of the bubbles, that causes the vibration and ultimate destruction of the impeller. It is definitely an unstable condition and should be avoided whenever possible.

Unfortunately, in centrifugals, the low pressures are caused by a combination of low gauge pressures and high velocities. The formation of the bubbles usually occurs near the impeller inlet and the downstream point at which they collapse depends on how far below the vapor pressure the local condition becomes. Movies of the Fairbanks-Morse pump test on one-half of a split case pump having a glass side, show that as the pressure in the pump becomes lower, the bubble lengthens and its collapse approaches the periphery of the impeller. As the pressure becomes low enough, the bubble will collapse outside the impeller in the discharge volute. It is evident that unequal distribution of velocities in the impeller passages also promotes cavitation. This condition always occurs at off peak capacities and often at the design point. Therefore, we conclude, the condition (capacity and head) at which cavitation begins cannot be predicted from the physical dimensions of the water passage. It must be determined empirically. Avoidance of high suction lift is paramount to successful pump operation. Our pumps are all tested at 15 foot suction lift which meets most water work specifications. The Hydraulic Institute recommends the maximum safe suction lift at various Specific Speeds for different impeller inlet types at the design point of impeller. It is based on the experience of the Institute membership. In event efficiency can be sacrificed, special large inlet impellers are designed for suction lifts higher than the Institute charts. This is common practice for hot water service in preference to lowering the pump to obtain the required head.

In general on a steep head-discharge characteristic for propeller pumps where change in discharge is small throughout the operating range, destructive cavitation usually occurs at maximum head condition. Just the opposite is true for impeller type pumps where cavitation occurs at the right end of the relatively flat characteristic head-discharge curve.

The pumps in between the low specific speed centrifugal and propeller pumps (the mixed flow) have a capacity range that is sensitive to suction lift near shut-off as well as at maximum capacities. On horizontal propeller pumps where the static lift is usually 8-12 feet, the pump may be quiet but is cavitating. The cavitation causes reduction in efficiency, but it it not destructive. I am showing you a sample of destructive effect of cavitation. It can be repaired by brazing, rebalanced and put back into service again.

As a summary on effects of cavitation in pump operation, there exists a tendency to operate low head, high capacity or high specific speed pumps close to the critical cavitating condition. Unfortunately, this is primarily due to economic consideration at the plant site.

DEEP WELL TURBINES

The deep well turbine pump and its allied vertical low lift pumps are essentially specialized adaptations of the conventional centrifugal type of pump. As in all centrifugals, the pressure-head against which the pump will operate depends on the diameter of the impeller and the speed at which it is rotated. In the deep well turbine pump the impellers are of limited diameter, hence the impeller is limited to the magnitude of head it will develop. By assembling a number of impellers with their surrounding cases (called stages in a series as multi-stage pump there is produced a pressure head which is in direct proportion to the number of stages. For example, if a single stage turbine pump will lift 500 gpm 40 feet @ a given rpm, ten stages of the same size will lift 500 gpm to a height (or head) of 400 feet.

In this turbine pumping section, the first impeller discharges into its case or "bowl" where the guide vanes and channels convert the velocity head into pressure head and then conduct the water under this increased pressure into the next impeller above. The next impeller adds to the water an identical quantity of energy, and in turn delivers it to the next impeller above, and so on. Identical quantity of water will be passing from stage to stage with each stage imparting an equal magnitude of pressure-head to it.

Performance characteristics depend upon design and construction. Because the pump is usually installed in a relatively small bore wall, it is obvious that in order to pump at its best performance for a certain range of capacity and discharge, the impeller must be designed accordingly. This means maximum quantity of water at greatest head with least horsepower.

Most turbine pump manufacturers now have a fully standardized line of turbine pump bowls and impellers which can be shipped from stock. These are standardized tested, and proven to fit almost every ordinary condition.

Deep well turbine pumps are usually designated by size; for example, 4", 6", 8" and 10" meaning the smallest actual inside diameter of well into which the pump can be installed. In order to allow for proper well clearance, the bowl or pump case O. D. is actually smaller than the nominal well bit and will range from 3-5/8" O.D. on 4" size pump to 23" O.D. on 24" size. Clearance is graduated according to size.

Pump impellers are designed also acording to relative capacity for a given diameter; for example, low, medium and high capacity. Hence a 12" low capacity is designated as 12" LC. A 12" medium capacity pump as 12" MC. An extra low or low capacity pump requires an impeller having relatively narrow passages and a large ratio between the inner diameter of the suction circle and the outer diameter would have inherently different characterstics; as compared with the high capacity pump having relatively wide radial channels and small ratio between the inlet circle and its outside diameter.

Usually a turbine pump is required to operate against conditions of varying heads, especially in deep well work where the water levels may vary considerably. In this case the downward sloping curve, or "steep characteristic", is most desirable. The horsepower will not vary to a great extent and it may be designed so that the maximum horsepower will be at the point of best performance of the pump. This is called a "non-overloading" characteristic. It is of great significance for diesel engine or gear head drives.

TURBINE SPECIFIC SPEEDS

Centrifugal Deep Well Pumps	Up to 4000
Mixflow Deep Well Pumps	Up to 4500
Niagara Mixflow (Figure 6360)	4000 to 9000
Nigara Propeller (Figure 6310)	9000 to 13000
Angle Flow Pump	4000 to 6000
Propeller or Screw	10000 — 15000
Axial Flow Pump	16000

Regardless of speed, the "peak" performance is usually about 60% of the shut-off head. The pump efficiencies vary between 70 and 90% depending on size, capacity and operating speed. For any particular head there is always a range of capacity for each speed (specific speed) that will cause lowest loss or given best efficiencies in a given pump. Proper selection is paramount.

Because the casing diameter limitation favours the axial impeller and column discharge, optimum efficiencies occur at mixed flow specific speeds up to 8000. However, at specific speeds over 4500, the BHP continues to increase as shut-off is approached from the design point. Normally, a mixed flow pump is never operated at shut off so that the size of the driver can be matched with the maximum operating head.

Besides selecting an efficient pump for the job, it is necessary to limit friction head added by the pump column or reduction pipe, oil tubing or shafting in case of a water lubricated pump. A compromise has to exist between cost of larger column and saving in horsepower, Saving of one horsepower per year at 2 per kilowatt hour is about $131.00 per year.

TYPES

Fairbanks, Morse & Co. manufacture both oil and water lubricated deep well turbine pumps. The water lubricated pump uses open-line shafting which is exposed to the water being pumped through the reduction column and shaft is supported by cutless rubber bearings. The bearings in the pump bowl are rubber also and are lubricated by the water being pumped.

The oil lubricated pump makes use of lineshafting which is encased in a cover tube permitting the use of metal bearings and oil as the lubricant. The top bowl bearing is also lubricated by oil as it acts as a terminus for the oil tube. The bottom bowl bearing is grease packed and the intermediate bowl bearings are open and exposed only to the water being pumped because the problem of applying individual oil lubrication to these bearings is extremely complex and costly.

It might appear that the more complicated construction necessary on the oil lubricated pump would make it the most costly to build. This is offset by the special materials and processing necessary to provide good non corrosive bearing surfaces which will not injure the rubber bearings, by the bearing retainer assemblies at each column joint, and by the special threads necessary to make up the column joints. Prejudices often affect the specifying of one type or the other. Many thousands of both types are in successful operation and both have their place in industry.

COMPARISON OF OIL AND WATER LUBRICATED PUMP REQUIREMENTS AS TO RIGHT ANGLE GEAR DRIVES AND BELTED TYPE HEAD

Figure 6970 *Open Line Shaft Water Lubricated*		*Figure* 6920 *Enclosed Line Shaft Oil Lubricated*
	Right Angle Gear Drives	
(*a*) Purchased by Branch from FM, Stuttgart, Ark.	—	same
(*b*) Requires adapter plate as it fits 6920 heads. Redesigned to V. H. S. Standard & & fit H6 Std.	—	Fits 6920 Std. head
(*c*) Must have adequate thrust capacity.	—	Same
(*d*) Must have non-reverse ratchet	—	Does not require nonreverse ratchet.
(*e*) Must be selected by power required, gear ratio, and correct direction of rotation of driver shaft.	—	same
	Belted Type Head	
(*a*) Purchased from Pomona, Calif. to fit HG heads.	—	Purchased from Pomona to fit 6920 heads.
(*b*) Selected by power, thrust and pulley size.	—	same.
(*c*) Pulley is replaceable in field as it is bolted to the pulley hub.	—	Pulley cast integrally with pulley hub and is not removable.
(*d*) Same as for geer drive. Need ratchet.	—	Same as for gear drive. Need no ratchet.

PRIME MOVERS

Probably the simplest, least expensive and most useful prime mover ever devised is the three phase squirrel cage induction motor. The vertical hollow shaft adaptation of this motor was developed for the deep well turbine industry. Mechanically it consists of the rotor or rotating element, which is mounted in a stationary housing which also supports the necesary thrust bearing, non-reverse ratchet (if such is required), radial bearing, lubricating system and mounting stand for connection to the motor pedestal on the pump head.

Other electric drives are:

2. Single-phase induction motor—Capacitor start, Induction run.
3. Single phase induction motor—Repulsion start, Induction run.
4. Wound rotor induction motor.
5. Synchronous motor.

It is possible to mount steam turbines vertically on the pump head. Vertical solid shaft electric motors, gasoline, diesel and natural gas engines and in rare instance specially built vertical internal combustion engines operate through the medium of a flexible coupling assembly which contains in addition to the flexible coupling a thrust bearing assembly and a means for adjusting the impeller clearance. Horizontal power units of all types apply their energy through the medium of belt and chain drives or right angle gear drives. Pulley ratios and gear drive ratios should be selected to make the pump operate as near the published conventional speeds as possible. If it is not possible, then performance corrections must be made.

Installation: The successful performance of a pumping unit depends primarily upon a good installation. No matter how well any pumping unit is designed and manufactured, poor installation will preclude rendering of satisfactory service. Printed instructions including procedure to be followed should be followed carefully. The Standards of the Hydraulic Institute as well as several operators' handbooks devote large sections to proper installation. Yet our letters of complaint indicate that instruction are often ignored.

Location: Pump should be installed as close to the water as practicable and preferably below the water level. The unit should be protected against flood and a motor-driven unit should be installed in a dry place. Provide adequate room for dismantling pump and inspection of its parts. Where considerable weight is involved on large pumps, head room must be made available for use of cranes or chain-blocks.

Foundation: This must be sufficiently substantial to absorb any vibration and provide rigid support for the bedplate. Concrete is the most satisfactory foundation. Bolts of the correct size should be located accurately and imbedded in the concrete through sleeves to permit some freedom of setting.

7. Discharge head too high, see item 5 above.
8. Air pockets in suction line, see discussion on piping.
9. Incorrect rotation, see item 1 above.
10. Mechanical defects, wearing rings are worn, impeller damaged from erosion, sleeves and packing worn. Inspect pump periodically and replace worn or defective parts.
11. Impeller diameter too small, check rating chart against total dynamic head on pump.

PREVENTIVE MAINTENANCE

Maintenance can for convenience be divided into two categories. One kind of maintenance is the repair of equipment which is broken down or not properly functioning. The other kind of maintenance is the "fix it before it breaks" or preventive type maintenance, which maintenance is designed to eliminate or minimize the necessity for break down maintenance. Your attention is directed to this discussion of preventive maintenance, or "fix it before it breaks", "a stitch in time saves nine", type of maintenance, as contrasted with break-down maintenance.

The value of eliminating or nearly eliminating the necessity for breakdown maintenance in public water utility is readily appreciated. Unless there is an over abundant amount of standby equipment installed, a water plant cannot let its major pieces of machinery run until they fail, then repair them. Equipment has the uncanny habit of breaking down just when it is needed most, or when it is most difficult to repair. Any program which will generally insure that equipment will continue to operate efficiently for a fairly definite future period is worth a great deal. Continuity of supply or disposal is one of the primary requisites of a pumping system. It is for this reason that we are to devote some attention to a preventive maintenance discussion, as the primary purpose of preventive maintenance is to prevent equipment outages and maintain original efficiencies and serviceability over the useful life of the equipment.

Preventive maintenance can also be termed "planned maintenance", or "scheduled maintenance." It is maintenance and care according to a pre-arranged plan and schedule, in order to keep service equipment and facilities in continuously proper operating conditions and repair.

BASIC PREVENTIVE MAINTENANCE

Preventive Maintenance in the water works consists of routine examinations, diagnosis and treatments. It consists of the following basic care operations applied to water works equipment and facilities.

First: Preventive Maintenance means the detection and correction of any unusual sound or change in appearance of equipment. For example, a knock in a gasoline motor, chattering in a centrifugal pump, thumping in an air compresser, or a squeak on a chemical feeder are often early signals of mechanical failure. Similarly, a wobbly "V" belt sheave, a chain drive riding high on the sprocket teeth, or gears not fully meshing are also indications of incipient mechanical failure requiring immediate attention. Obviously, the water works operator should correct or report any such changes that he notices. On occasion, the mere critical eye or ear of a competent mechanic is required to detect operating irregularities which an operator will miss. *When corrected,* preventive maintenance has been practiced.

Second: Preventive Maintenance is keeping equipment clean. Moisture, dirt, dust, cobwebs, bugs, oil in the wrong places are definitely enemies of mechanical and electrical equipment. Bird nests in out-door switching gear and motors, bugs in lubricating oil and dust in motors may cause failures that would be prevented when equipment is kept clean.

A vacuum type cleaner and a compressed air line for blowing out equipment are almost "musts" in keeping equipment clean. Then too, if equipment is to be kept clean, the surroundings must similarly be kept clean. Mopping of the floors, removal of debris from buildings, cleaning of cobwebs from coilings, elimination of oil drips, or general "good housekeeping", is an important part of keeping equipment clean in preventive maintenance.

Third: Preventive Maintenance means that equipment must be protected against external or environmental conditions which cause general deterioration or disintegration. For example, open motors should not be used in extreme dusty atmospheres, machinery requiring the dissipation of generated heat should not be installed where proper cooling cannot be applied. Equipment which would be harmed by water should not be located near potentially leaking pipes, tanks and fixtures. Thus, the operator or the mechanic concerned with preventive maintenance must be able to recognize conditions external to the equipment that would cause its deterioration, such as improperly placed lockers cutting off ventilation or an incorrect use of a water hose around motors, or the use of an air hammer or paint spray near open bearings or motors.

Fourth: Preventive Maintenance is keeping equipment in a state of "good order". For example, metal subject to corrosion must be kept painted or covered with a protective covering, loose nuts and bolts should be tightened, leaning power poles should be straightened, sagging guy wires should be put in proper tension, leaking roofs should be repaired even though it isn't raining. Those too are "musts" in preventive maintenance.

Fifth: Preventive Maintenance means the correction of conditions which cause heat in excess of that for which the equipment was designed. Excessive heat may be the result of any number of incipient mechanical failures, from improper lubrication or improper fit, to overload. In many cases, "excessive heat" must be defined from manufacturer's recommendations or other experience records. Thermometers or surface pyrometers often must be used. Two different

operators may not have the same touch sensitivity, ortwo different bearings may have decidedly different heat tolerances.

Sixth: Preventive Maintenance means the elimination of excessive vibration, Vibration, or vibration tolerances, must in some cases be defined and a vibration motor used to determine the magnitude. A 10 mil vibration* for some equipment is excessive and for another kind of equipment is well within safe limits. Preventive maintenance requires an understanding of "what is vibration".

Vibration may be caused by misalignment, out-of-balance, loose nuts and bolts, structural breaks. When vibration does occur, all contributing causes must be carefully checked.

Maintenance of both static and dynamic balances** of such rotating parts as pump impellers and motor rotors is especially important in the prevention of vibration. Some phases of preventive maintenance require routine dynamic balance checks in addition to static balance checks. This is specially true in pumps with overhanging solids handling impellers which are apt to become partially plugged.

Seventh: Preventive Maintenance is the periodic testing for proper operation. Mechanical trouble is often discovered by a testing procedure which shows the equipment to be incapable of full capacity performance, operating inefficiently, or which shows the equipment to be operating under conditions for which it was not designed.

Such testing procedures may be relatively simple but nevertheless important in preventing operation a failures. A chlorinator or a gasoline engine may be apparently operating satisfactorily at one-fourth capacity rating but trouble is discovered when given a full capacity operating test. A pump efficiency test which may merely require the reading of existing motors and the calculation of efficiency, might indicate worn parts, a collapsed suction strainer, a sticking check valve.

Other tests might require more time and trouble, but also are properly a part of any preventive maintence program. A drop test on a filter or a basin to determine proper motor registration or a line slaker heat balance test might be a clase in point.

Eighth: Preventive Maintenance is the proper lubricatio n of equipment, which means the proper lubricant, properly applied at proper intervals. The dangers of over-lubrication are as important as under-lubrication.

Effective preventive maintenance requires a carefully worked out lubrication plan. As an example, different pressure grease fittings could be used for each different lubrication grease to prevent misapplication of grease lubricants. Or a color and symbol scheme might be used to mark both the points of lubrication and the container or grease guns storing the lubricant. The misapplication of lubricants is a common occurence and must be preverted.

Ninth: Preventive Maintenance is the pericdic check for wear and repair or replacement before it beccmes excessive. Such inspections often require the complete disassembling and reassembling of the equipment.

Centrifugal pump, for example, may be given an annual overhaul, based upon hours of service. in which bearings and ring clearances are restored, sleeves are rebuilt, casings are painted, impellers are balanced when necessary. This should result in continued pumping service without breakdowns or loss in efficiency. Chemical feeds, motors, rate-of-flow controllers, should be similarly over-hauled on a periodic basis in order to discover worn parts and to prevent breakdowns.

The above nine points are basic care operations in preventive maintenance. Both the Doctor and the Engineer first give their patients a thorough physical examination. Then the Doctor measures heart beat, and the Engineer measures water, oil and gass pressures. Both the Doctor and the Engineer measure temperatures and misalignments. The Doctor makes his basic metablolism test and the Engineer his fuel or power consumed test. The Doctor tests for valve action on the electro-cardiograph and the Engineer makes indicator cards. The ccmparison can be expanded, but in one respect the Engineer has a decided advantage over the Doctor—he can easily take his patients apart to inspect and correct for internal wear.

Somewhere in this discussion, your attention should also be directed to a rather obvious fact. The practice of preventive maintenance requires that the necessary tools and skills are available. Yet many plants do not possess even such simple tools as feeler gauges, micrometors, thermometers, adequate wrench sets, good paint brushes, etc. In large plants, presses, electric and gas welding and cutting outfits, metalizing equipment, sand blast equipment, tong ammeters, reccrding ammeters, volt motors, thermometers and pressure gauges, indicators, techmeters, and similar testing and maintenance tools may be needed. When plant personnel does not include men with necessary skills, or where costly items of equipment are not used often enough to warrant the investment necessary to obtain them, arrangements should be made for hiring necessary specialists, and renting the necessary equip ment and tools.

As indicated in the opening paragraphs, Preventive Maintenance consisting of these basic care operations must be done regularly according to a pre-arranged schedule. This means that such work as lubrications, clean-ups, and inspections must be done routinely and must not be subject to a hit and miss system, such as: to be done as the operator has time in between his

* R.T. Kent—Kent's Mechanical Engineers' Handbook III—1936—P 16-04

** R.T. Rent—Rent's Mechanical Engineers' Handbook II—1945—P 8-38

other duties. The same attention and importance must be attached to preventive maintenance as is given to washing of filters, adding of coagulants to the water, or starting and stopping of pumps. Unless preventive maintenance is regarded as an essential operational procedure, *it cannot be effective.*

Permit me a few illustrations. If experience indicates it is desirable to blow out an electric motor on a chemical conveyor at least once each week, then this clean-up operation must be performed routinely, without miss, on a weekly basis if the performance life of the motor is to be preserved. To miss one week is to invite a second miss, and a third, unless—and who can tell how long a dirty motor will operate.

Or, perhaps, it has been found that a chemical solution tanks should be drained and the lining carefully inspected for flaws and pits every six months. The results of having overlooked such an inspection, because the operator was very busy, may or may not be tank failure. However, there is no reasonable assurance that the tank will remain sound during the peak pumping period unless inspections are made routinely and on time.

Again, let us assume that a manufacturer has recommended the lubrication of a bearing every six months. Experience has confirmed the reasonableness of this recommendation. Good judgment dictates that this lubrication be scheduled routinely as recommended, and not just when the operator happens to remember to do it. Anything short of this might invite an unscheduled bearing failure.

To say that unless the utility religously follows a scheduled maintenance program, breakdowns and interruptions in service are inevitable, may be stretching the truth. But, on the other hand, it can be said without reservation, that unless preventive maintenance is scheduled and routinely performed *on time*, there is no assurance that such break-down will not occur.

This brings us to our next consideration. Scheduled preventive maintenance must be resolved into a *written program*. That is, preventive maintenance planning and calendaring is only effectively carried out when it becomes a written record and program. It is extremely unwise to keep the answers to such questions as—when should the oil in the circuit breaker again be sampled and sent out for test; what preventive maintenance is required for No. 1 Chemical Feeder; what kind of lubricant is recommended for that thrust bearing on scraps of paper or in the memories of the superintendent or plant operator Such practice would be comparable to retaining the locations of valves and mains in the distribution system in the minds of one or two old timers in the distribution department. Instead, a carefully written system of records and maintenance schedules are necessary.

Preventive Maintenance operations are too numerous even in the small water utility to depend upon the operators' or superintendents' memories as to when and how they are to be done. A *written schedule* is necessary to show when preventive maintenance is to be done, how it is to be performed, who is to perform it, when it was last performed, and the condition of equipment as of the previous inspection. Then too, the thoroughness of preventive maintenance planning is probably not fully known until it is charted on paper. As a Professor of English, often told his classes, you are not certain of the extent of your knowledge regarding a subject until you attempt to express it in language.

Up to this point, an attempt has been to show that the purpose of preventive maintenance is to prevent interruptions or the curtailing of utility service. Then preventive maintenance was described as having some similarity to preventive medicine, and that it consists of several basic care procedures. Next, it was pointed out that effective preventive maintenance must be planned and scheduled, and that such planing must be *written* (i.e., charted) before it can effectively be put into practice.

This practice of preventive maintenance calls for setting up certain record and report *forms*, and *routines* for handling them. The particular kind of forms and the kind of routines are relatively unimportant, providing of course that they properly assist in the practice of preventive maintenance. Readers of such magazines as McGraw-Hill's "Factory Management" are aware of the number of articles which have been published showing the forms and routines in use in various industries. Collectively, these articles indicate that there are several types of preventive maintenance record forms and routines in use in industry, each adapted to a particular company's needs. Similarly, in a public utility preventive maintenance record forms must be adapted to the utility's need.

Even though the preventive maintenance record and procedure system for public water utility need not follow a standard pattern, they should at least have the following general characteristics:

(1) The record system and detail should not be cumbersome and burdensome—that is, the detail and clerical work required by record forms should consume a minimum of time. Obviously, the important consideration is the preventive maintenance to be done and not the record system. It would be possible but unwise, to devise a record system that could easily require more time than the preventive maintenance which it represents.

(2) The records and forms should list the planned preventive maintenance operations for equipment; such as, the necessary inspections, lubrications and replacements, on a definite pericdicbasis. This, obviously, is the primary function of the record system. That is, through the use of the preventive maintenance records, the Superintendent or operator of the utility should readily be able to tell "what preventive maintenance is scheduled", "at what intervals it is scheduled", and "when it is to be done", for any

equipment. For example, the records should show perhaps that the filter rate of flow gauges should be checked by filter drop test four (4) times each year, during the first week of January, April, July and October.

(3) Any portion of the record system should be *adaptable to easy evision* often and whenever such revision is desirable, in order to keep the record system continuously up-to date. The amount and kind of preventive maintenance, its frequency, for any equipment, often requires revision whenever experience or changes in operating conditions dictate that such changes are desirable. The kind and amount of preventive maintenance required by equipment is never a static thing. First judgment as to the kind of preventive maintenance required are often wrong. Then as equipment grows older or operating conditions change, the frequency for making certain inspections should be increased or decreased. It is essential that these changes can be easily indicated in the records without making other portions of the records obsolete. Revisions to a record system are often most easily made when the record system is of the loose leaf or catalogue file type.

(4) The record system should also be a histroical record of the physical condition of the equipment. A continuous periodic study of inspection reports will often indicate such things as rate of obsolescence of equipment, rate of wear and other conditions which are causing equipment destruction. With some equipment, unless comparisons with previous inspections are made, much of the value of preventive maintenance inspection are lost.

These are perhaps the general essentials of a preventive maintenance record system. The record system could also properly be designed to be a part of cost accounting, stores records, or work assignment forms. Obviously, the record system must be designed to fit the needs and the available personnel in the utility.

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Pump Input

List of Symbols:

$C = g H_2$
d = Inlet dia. of impeller
d_1 = Output dia. of impeller
H_1 = Input head based on outlet whirl (effective value)
H_2 = Subtractive term in input head based on inlet whirl (effective value)
K = Short circuit leakage coefficient $= \frac{Q_t}{Q}$
$K' = \frac{1}{g}$ where g is gravitational acceleration
P_o = disc friction plus stuffing box and journal friction loss
P_t = Pump input
$\overline{P_t}$ = Pump input at designed point
$P_{t\ max}$ = Max. pump input
Q = Rate of out-flow from the pump
$\overline{Q}$ = The flow from pump at designed points
Q_t = Q + Short-circuit-leakage
$Q_{t\ max}$ = Quantity at which max pump input occurs.
V_{f1} = Velocity of flow at outlet at design point
V_{we1} = Effective outlet whirl
V_{we} = Effective inlet whirl
V_1 = Outlet peripheral velocity of impeller
V = Inlet peripheral velocity of impeller
Φ_e = Effective outlet angle of vanes.

The design of Centrifugal pumps can be as simple or as difficult as the standard of performance aimed at. Amongst the requirements which reduce the discrepancy between theory and practice and bring the performance of pump under control, the main ones are:

(*a*) Effective value of outlet whirl for given impeller dimensions, vane curvature and number of vanes.

(*b*) Existence, control and computation of inlet whirl (effective value) for given parameters as above, as well as for given impeller and shaft hub diameters.

(*c*) Determination of vane curvature and number of vanes from all types of impellers.

(*d*) Calculation of hydraulic and mechanical losses of pump so as to arrive at the output head and efficiency.

(*e*) Calculation of pump input, maximum pump input and the capacity at which this occurs.

It is with the last aspect (*e*) that this paper is concerned, although on each of the above I have done original work and proved it in practice. In the design of Centrifugal and Axial flow pumps, it is not only necessary to achieve the required range at the required head but also to obtain these with minimum pump input so as to attain a high efficiency. It is also not sufficient to know the pump input at the design point but to so arrange that the maximum input anywhere throughout the range does not much differ from the input at the designed point. If this condition is not attempted an unnecessary power will have to be tied up in the driver thus increasing the initial outlay and also reducing the overall efficiency as the driver at partial loads is likely to have lesser efficiency.

By M. W. LALCHANDANI

Aspects of Pump Input

Hence it is useful to know the following during the design of a pump:—

(1) The pump input at the design point
(2) The maximum pump input
(3) The capacity at which the max. pump input would occur.

The knowledge of the above would help to achieve a minimum possible 'max. pump input' by adjusting the various pump dimensions. The word 'possible' is used since the manufacturing and performance considerations impose certain limitations on the pump dimensions. The following analysis has been derived and used by me in several designs of varying nature and the actual results have proved to conform to the esxpected data.

Maximum Pump Input

The pump input is made up of the hydraulic input and the mechanical input. The hydraulic input depends on the capacity being pumped out, the amount of short circuit leakage loss and the input head. The mechanical input covers (*i*) disc friction of impeller and (*ii*) stuffing box, journal and any other mechanical loss.

Thus $P_t = Q_t (H_1 - H_2) + P_o$ (1)

where P_t = Pump Input i.e. total power

$K = \frac{Q_t}{Q}$ the leakage coefficient

$H_1 - H_2$ = Input head depending on outlet and inlet effective whirls.

P_o = disc friction and mechanical losses

Expanding:

$$P_t = Q_t \left(\frac{V_{we} v_1}{g} - \frac{V_{we1} v}{g} \right) + p_o \qquad (2)$$

$$= K' Q_t \left\{ (v_1^2 - v_1 V_{f1} \cot \Phi_e \frac{Qt}{\overline{Qt}} - C \right\} + p_o \qquad (3)$$

$$\text{where } K' = \frac{1}{g}$$

H_2 depending on inlet whirl V_{we} having a smaller value and changing but little from Qt to max. is considered a constant C in order to simplify the analysis, the simplification not distorting the result as is borne out in practice.

To Know at which Capacity the maximum Pump input occurs

P_t may be differentiated with respect to Qt and quated to zero.

$$\frac{dP}{dQ_t}^t = K'(v_1^2 - 2v_1 V_{f1} \cot \Phi_e \frac{Q_t}{\overline{Q_t}} - C) = O \qquad 4$$

$$Qt_{max} = \overline{Qt} \frac{v_1^2 - C)}{2v_1 \overline{Vf_1 \cot \Phi_e}} \qquad 5$$

$$= \overline{Q_t} \frac{v_1^2 - gh_2)}{2v_1 Vf_1 \cot \Phi_e} \qquad 6$$

$$Q_m = \frac{1}{K} Qt_{max} \qquad 7$$

Where Q_m is the capacity from pump at which max. pump input occurs.

Substituting Qt_{max} in P_t

$$P_{tmax} = K' \overline{Qt} \frac{v_1^2 - C}{2v_1 Vf_1 \cot \Phi_e} \left(v_1^2 - \frac{(v_1^2 - C)}{2} - C \right) + p_o$$

$$= K' \overline{Qt} \frac{(v_1^2 - C)}{2v_1 Vf_1 \cot \Phi_e} \frac{(v_1^2 - C)}{2} + p_o$$

$$= K' \overline{Qt} \frac{(v_1^2 - C)^2}{4v_1 Vf_1 \cot \Phi_e} + p_o \qquad (8)$$

Thus equation (8) gives the max. pum input and (6) the capacity at which this occurs.

To adjust max input at designed point:

To know what relationship must exist between V_1, Vf_1 and cot Φ_e in order to bring about the max. pump input at designed point, $Pt_{max} = \overline{P}_t$

$$\therefore K' \overline{Qt}(v_1^2 - v_1 Vf_1 \cot \Phi_e - C) + p_o$$

$$= K' \overline{Qt} \frac{(v_1^2 - C)^2}{4v_1 Vf_1 \cot \Phi_e} + p_o \qquad (9)$$

$$\therefore v_1^2 - v_1 Vf_1 \cot \Phi_e - C = \frac{(v_1^2 - C)^2}{4v_1 Vf_1 \cot \Phi_e}$$

$$\therefore A - B \cot \Phi_e = D \tan \Phi_e$$

Where $A = v_1^2 - C$

$B = v_1 Vf_1$

$D = \frac{v_1^2 - C)^2}{4v_1 Vf_1}$

$$\therefore A \tan \Phi_e - B = D \tan^2 \Phi_e$$

$$\therefore D \tan^2 \Phi_e - A \tan \Phi_e + B = C$$

$$\therefore \tan \Phi_e = \frac{A \pm \sqrt{A^2 - 4DB}}{2D}$$

$$= \frac{2v_1 Vf_1 [(v_1^2 - C) \pm \sqrt{(v_1^2 - C)^2 - (v_1^2 - C)}}{(v_1^2 - C)^2}$$

$$= \frac{2v_1 Vf_1}{v_1^2 - C} \qquad \ldots (10)$$

If C is neglected,

$$\tan \Phi_e = \frac{2v_1 Vf_1}{v_1^2} \qquad \ldots (11)$$

$$\therefore Vf_1 \cot \Phi_e = \frac{V_1}{2} \qquad (12)$$

$$\therefore v_1 - V_{we\,1} = \frac{V_1}{2}$$

$$\therefore V_{we\,1} = \frac{v_1}{2} \qquad (13)$$

If C is not neglected, from (10)

$$Vf_1 \cot \Phi_e = \frac{v_1^2 - C}{2v_1}$$

$$= \frac{v_1}{2} - \frac{C}{2v_1}$$

$$= \frac{v_1}{2} - \frac{gH_2}{2v_1}$$

$$= \frac{v_1}{2} - \frac{vV_{we}}{2v_1}$$

$$= \frac{v_1}{2} - \frac{d}{2d_1} V_{we} \qquad .. \qquad (14)$$

i.e . $Vf_1 \cot \Phi_e$ is less than half of outlet peripheral velocity by the factor $\frac{d}{2d_1} V_{we}$

Also $v_1 - V_{we1} = \frac{v_1}{2} - \frac{dV_{we}}{2d_1}$

$$\therefore V_{we1} = \frac{v_1}{2} + \frac{dV_{we}}{2d_1} \qquad .. \qquad (15)$$

Pump Input

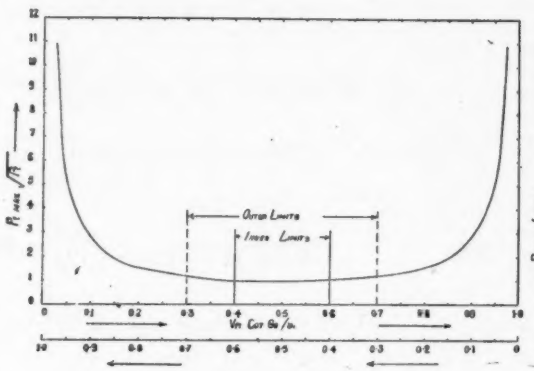


Fig 1.

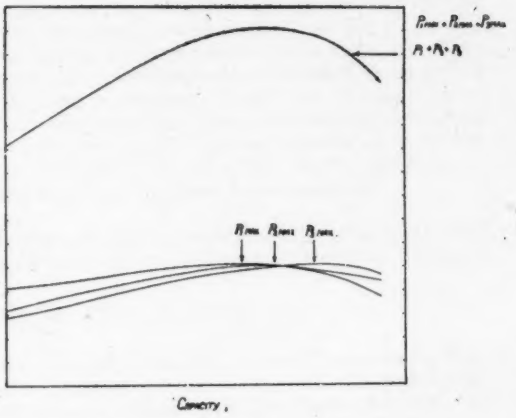


Fig 2.

i. e. V_{we1} is more than half the outlet peripheral velocity by the factor $\frac{d}{2d_1}$ V_{we}.

Ratio of max. input to design point input

Thus whenever $Vf_1 \cot \Phi_e$ or V_{we} are other than the values given by)14) and (15), Pt_{max} will occur at other than the designed point Q.

$$\frac{Pt_{max}}{Pt} = \frac{K' Qt \frac{(v_1^2 - C)^2}{4v_1 Vf_1 \cot \Phi_e} + p_o}{K' Qt (v_1^2 - v_1 Vf_1 \cot \Phi_e - C) + p_o}$$

Neglecting P_o since it is small in most of the case,

$$\frac{Pt_{max}}{Pt} = \frac{(v_1^2 - C)^2}{4v_1 Vf_1 \cot \Phi_e (v_1^2 - v_1 Vf_1 \cot \Phi_e - C)}$$

Since C is also small compared to v_1^2, neglecting it for simplicity,

$$\frac{Pt_{max}}{Pt} = \frac{v_1^2}{4Vf_1 \cot \Phi_e (v_1 - Vf_1 \cot \Phi_e} \quad (16)$$

$$= (Vf_1 \cot \Phi_e + V_{we})^2 / 4Vf_1 \cot \Phi_e . V_{we}$$

$$= \frac{(vf_1 \cot \Phi_e + V_{we})^2}{Vf_1 \cot \Phi_e + V_{we1})^2 - (Vf_1 \cot \Phi_e - V_{we1})^2}$$

$$= 1 + \frac{(vf_1 \cot \Phi_e \sim V_{we1})^2}{4Vf_1 \cot \Phi_e V_{we1}} \quad (17)$$

The ratio $\frac{Pt_{max}}{Pt}$ is in excess of unity by an amount which varies as the square of difference between $Vf_1 \cot \Phi_e$ and V_{we1}. If this difference is small, the square will be still smaller and this divided by $4Vf_1 \cot \Phi_e$. V_{we} would make this excessover unity negligible. Hence the max. pump input would differ only slightly from the pump input at the designed point when V_{we} does not differ much from $Vf_1 \cot \Phi_e$ i.e.

When V_{we1} is somewhat more than $\frac{V_1}{2}$

or $Vf_1 \cot \Phi_e$ is somewhat less than $\frac{V_1}{2}$

But if for example $V_{we1} = \frac{3}{4} v_1$

$$\therefore \quad Vf_1 \cot \Phi_e = \tfrac{1}{4} v_1$$

$$\therefore \quad \text{Ratio} = 1 + \frac{(\frac{1}{2} v_1)^2}{4 \times \frac{3}{16} v_1^2}$$

$$= 1 + \tfrac{1}{4} / \tfrac{3}{4}$$

$$= 1.33$$

The graph Fig. 1 gives the power ratio (max. pump input to input at the designed point) against $Vf_1 \cot \Phi_e$ & V_{we1} expressed as fractions of outlet peripheral velocity.

Design Procedure

Pump input at designed point depends upon input head $(H_1 - H_2)$ and total flow Qt together with the disc friction and mechanical losses like stuffing box and journal friction. Input head is to be such that when the hydraulic losses are deducted from it, the required output head is obtained. In other words for a given output head there has to be a certain input head and this is to be prejudged on the assumption of hydraulic losses. The parameters v_1, Vf_1, $\cot \Phi_e$ are to be so chosen that they bear out the assumptions.

Hence v_1 is first selected from the consideration of shut off head and therefore on the outside diameter of impeller. $Vf_1 \cot \Phi_e$ follows from $v_1 - V_{we1}$ as $\frac{gH_1}{v_1} = V_{we}$ for given $Vf_1 \cot \Phi_e$, Vf_1 and $\cot \Phi_e$ can be a shuffled so as to obtain as wide an outlet width of impeller, and as large an outlet angle (efffective angle Φ_e). But the last two requirements are opposed to each other as although manufacturing and performance requirements desire this, the given value of $Vf_1 \cot \Phi_e$ makes a large Vf_1 or small outlet width associated with smaller $\cot \Phi_e$ i.e. large Φ_e.

Thus the design of a centrifugal (or Axial pump) is a highly studied compromise between various conflicting requirements as it has to satisfy to a maximum possible extent the manufacturing limitations or desideratum on one hand and the desirable features of high efficiency low pump input, and the maintenance of these qualities over a wide range of head and quantity so as to enlarge the scope of a given pump.

Procedure for Francis, Mixed flow and Axial flow Impellers

The max. pump input and the flow at which this occurs can be calculated at the design stage for each streamline (the flow is divided by number of stream tubes) and the average P_{max} computed. But care must be taken that Qt for each stream line is nearly same as otherwise the average P_{max} cannot truely represent the max. pump input as shown in Fig. 2. This can be done by shuffling up the values of v_1, Vf_1, and Φ_e.

Some Studies on Air-Entraining Vortices in Pump Sumps

By N. S. LAKSHMANA RAO

It is generally recognised and accepted that the performance of a pump is greatly affected by the design of the sump or suction-well from which water is drawn. Instances are reported of rough running and overloading of pumps due to faulty sump design. The design of the sump assumes added importance in the light of modern trends towards adoption of high specific speed centrifugal pumps which are susceptible to flow conditions at intake structures.

Air entrainment occurs through vortices which are formed due to inadequate submergence of the suction pipe below the free surface of water in the sump. When the pump delivers an air-water mixture instead of water it is obvious that its performance is considerably affected and delivery lowered. The unstable vortices that are formed give rise to unsteady swirl flows in the pipe which result in fluctuating loads on pump bearings[12]. It would be expensive, indeed, to excavate sumps deep enough to ensure safe submergence.

Care must be taken to prevent the formation of air entraining vortices and swirl in the design of intakes to a turbine or pump. The air carried into an intake by a vortex may easily amount to 5% of water flow which will have disastrous effects on the efficiency of hydraulic machinery. Serious drops in pump efficiency of hydraulic machinery. Serious drops in pump efficiency may be caused by swirling flow if intakes are short. The basic cause of air-entraining vortices can be attributed to the presence of persistent rotational flow in the body of water approaching the intakes. The location of the intakes relative to the direction and boundaries of the approaching flow decides the strength of this rotational flow. The British Hydromechanics Research Association, [13] a pioneer in this field of study, has made 1/30 scale model studies of intake structures of the Ffestiniog pumped-storage scheme to find out the conditions causing air-entraining vortices when the reservoir is drawn down through the normal daily range of 65 ft. and has suggested various remedies.

The objects of the investigations reported in this paper are to find out the conditions under which different vortices form, the study the effect of different submergences on pump performance under different sump conditions and to suggest a simple, economical and effective sump geometry together with an inexpensive flat-finned conical fixture to the suction pipe to ensure an efficient pump performance by preventing the formation of vortices.

REVIEW OF EXISTING LITERATURE

Published information on the exact conditions under which the vortices are formed is somewhat contradictory.

Regarding the cause of vortices, Addison (1948) mentions that the swirl round corners or obstructions may set up vortex-forming conditions. Stepanoff (1948) on the other hand, states that all vortices originate in the impeller, which is not true always as pre-rotation does not travel far upstream at normal pump deliveries. Dominion Engineer (1948) gives little definite information beyond stating that "any kind of whirl conditions at the inlet to the impeller rediuces the capacity of the pump and may cause cavitation and noisy operation together with reduction in efficiency." Worster (1949) concludes that the geometry of the sump, location of the suction pipe and bellmouth and Froude number of flow are the controlling factors of vortex formation. These conclusions of Worster are in conformity with Binnie's (1948) work on inverse case of a bell-mouth spillway where the mouth points upwards. Waldo (1950) takes the factors involved in vortex formation as either due to rotation of the impeller or due to the flow in the suction pipe. In a graphical representation of peripheral speed at the inlet blade edges versus suction pipe velocity, he has shown the safe submergence values for given cases. Denny (1952) states that "regarding flow in the suction pipe, it was noticed that rotation of the water in the pipe was related to the swirl in the tank and seemed unaffected by whether or not a vortex existed." Young (1952) observes that pre-rotation of flow in the suction pipe occurs for all discharges below normal and that no back flow or swirl is found at high deliveries. Iverson (1953) appears nearer the truth when he states "while this effect (pre-rotation of the fluid entering the suction-bell as a result of impeller action) is undoubtedly present, the major influence stems....from the sump design and the flow pattern in the sump". Fraser (1953) attributes vortex formation to the intake design which causes a change in the direction of flow before the water enters the suction pipe. Denny (1955) writes that vortices are caused by rotation of the mass of water within the sump arising primarily from entry conditions. Their severity, measured in terms of the minimum safe submergence to avoid air-entrainment was also found to depend largely on the velocity in the suction-inlet; air-entrainment rarely occured below a velocity of 2ft/sec., and above 15 or 20ft/sec., the severity of the vortices did not increase substantially. Denny concludes that although the mechanisms of vortex formation and air-entrainment are complex, they do show regular and repeatable behaviour. Vortices are most severe when the swirl in the sump is considerable or when the pipe in located at the centre of the swirl but, at the other extreme, vortices can still form with imperceptible swirl in the sump. The likelihood of vortices is increased by

Some Studies on Air-Entraining Vortices in Pump Sumps

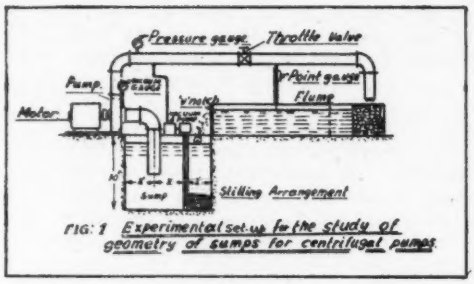


FIG: 1 Experimental set-up for the study of geometry of sumps for centrifugal pumps.

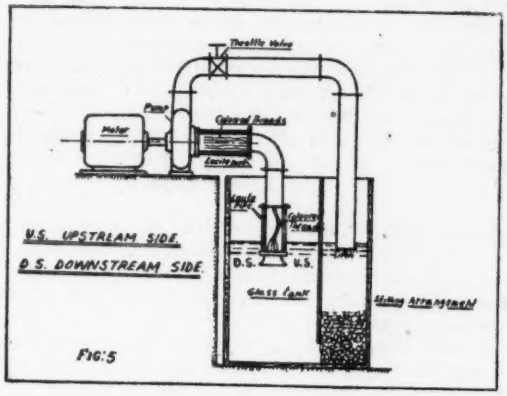


FIG: 5

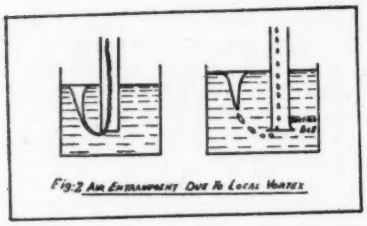
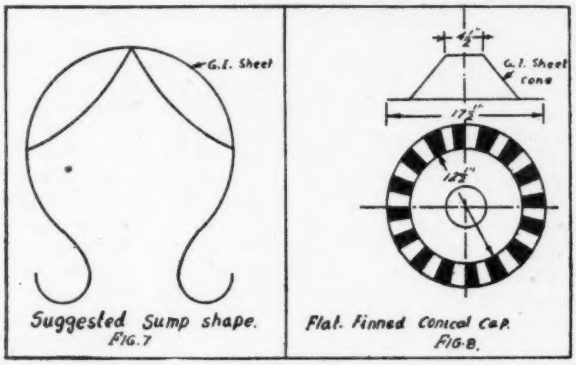


FIG: 2 AIR ENTRAINMENT DUE TO LOCAL VORTEX



Suggested Sump shape. FIG. 7

Flat. Finned Conical Cap. FIG. 8.

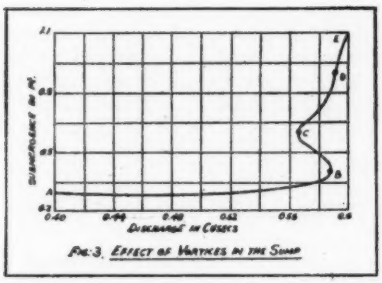


FIG: 3. EFFECT OF VORTICES IN THE SUMP

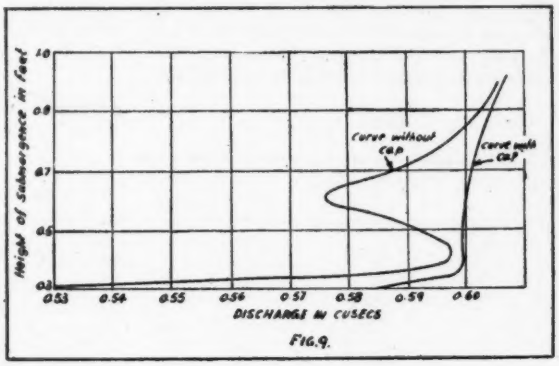


FIG. 9.

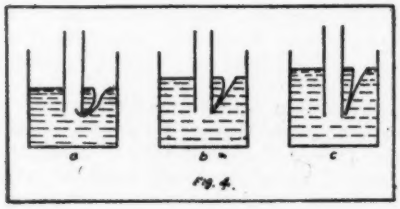


Fig. 4.

P20

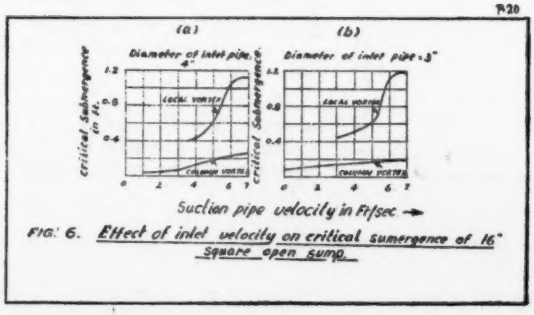


FIG: 6. Effect of inlet velocity on critical submergence of 16" square open sump.

asymmetrical lay-outs and sudden changes in the width or direction of the flow channel that cause large swirls in the sump. Sometimes the practice of constructing the sump to the dimensions demanded by the machinery in the pumphouse leads to sumps that are larger than desirable and frequently to sudden enlargements of flow channels which are a potential source of vortices.

Regarding the prevention of vortices, Stepanoff (1948) claims that a submergence of 5 feet on the suction bell edge is sufficient, but with low velocities in the inlet (2-3 ft./sec.), a submergence of half a diameter is safe. Dominion Engineer (1948) recommends that it is often good practice for compact lay-outs to use direct connected suction elbows of special design to obtain true axial flow in the impeller. These elbows are somewhat similar to the elbow type draft tube of hydraulic turbines, being arranged, however, to suit reverse flow direction. The cross sections of these elbows provide for a steady increase of velocity and are shaped to prevent whirl. Dominion Engineer further suggests a low roof over the sump as the best safeguard against whirlpools as it would minimise the air space between it and the water surface. Another remedy would be a shield below the water surface of such dimensions that the distance would be too long for whirls connecting the water surface with the intake. Waltman (1950) suggests that the bell-mouth should be submerged to an amount equal to three times the velocity-head plus one foot. Denny (1955) observes that the shape and disposition of the suction inlet only slightly affect the vortices, but the sump boundaries considerably reduce vortex formation when closer than 8 diameters to the suction pipe. With the walls very close to the pipe, vortices do not occur. Small sumps may thus be more satisfactory than spacious ones.

STUDIES AT THE INDIAN INSTITUTE OF SCIENCE

Experimental Set-up:

The set-up (Fig. 1) consisted of a suction sump 4 ft. by 6 ft. and 10 ft. deep with a vertical suction pipe leading to a centrifugal pump driven by a 7.5 H.P. motor, having its delivery valve fully open. The flow over the notch was measured accurately by using an electronic point gauge. The water, after measurement, was allowed to flow back into the sump through a stilling arrangement. (Fig. 2)

Tests were carried out starting from a minimum value of submergence. Submergence is defined as the depth of the lowest point of the suction pipe below the general water level in the sump. The submergence was increased to various desired values by means of a second pump which was independent of the main set-up. The effect of vortex formation on pump discharge under different sump conditions was studied.

Experimental results:

Suction failed at initial values of submergence. With the gradual increase in submergence, suction started and water commenced to swirl round the suction pipe causing a draw-down around the periphery of the column. This drawdown would cause a momentary drop in the water level and air would be sucked in for a few seconds lowering the delivery head. The reduced delivery into the flume would increase the water level in the sump and cause a smaller draw-down in it. This type of vortex around the periphery of the suction column may be called as 'Column Vortex'.

The effect of Column Vortex was observed to decrease with increase in submergence. A vortex would now form away from the suction pipe and when it is drawn towards the suction pipe, air-entrainment would occur with considerable noise. This type of vortex may be termed a 'Local Vortex'. This occured when an eddy reached regions of higher velocity near the suction bell. A close observation of the air-core being sucked in indicated that it was preceded by a flow of individual bubbles (Fig. 2).

To start with, vortex formation would be closer to the pipe, tending to be less stable and would usually revolve slowly round the pipe and then disappear; by this time another vortex would form and the cycle of events would repeat itself. With the increase in submergence, a local vortex would form away at a distance from the suction pipe with the core lengthened to reach the mouth of the suction pipe. The local vortex was found to be more stable and predominant upto a certain critical value of submergence, then started decreasing and with further increase in submergence, it completely disappeared.

For various values of submergence, readings were noted for critical flow during the formation of air-entraining vortices (Table I) and a curve of discharge versus submergence was drawn (Fig. 3). The total head was maintained constant in all the experiments.

Table I

Sl. No.	*Submergence in Ft.*	*Discharge in Cusecs.*
1.	0.22	0.3030
2.	0.25	0.2067
3.	0.50	0.4375
4.	0.32	0.5632
5.	0.35	0.5837
6.	0.40	0.5887
7.	0.45	0.5867
8.	0.50	0.5783
9.	0.55	0.5709
10.	0.60	0.5660
11.	0.65	0.5672
12.	0.70	0.5790
13.	0.75	0.5867
14.	0.80	0.5892
15.	0.85	0.5892
16.	0.90	0.5902
17.	0.95	0.5920
18.	1.00	0.5945
19.	1.05	0.5972

DISCUSSION OF EXPERIMENTAL RESULTS

(*i*) Behaviour of Vortices:—

As submergence increases a steep rise in flow can be noticed from A to B (Fig. ?). With further increase in submergence, the air-entrainment due to the draw down around the periphery of the suction pipe is reduced as the same would be checked by the suction pipe walls.

Besides air entrainment other effects are also caused. For instance, as the flow is very unstable, the periodic flow produces velocity fluctuations with corresponding fluctuations in the impeller thrust. The effect of local vortex is not present at this state of low submergence (A to B in Fig. 3) as the depth and hence the strength of eddies is small making the ability to such air, weak. Also the vorter will have to suffer a serious distortion in bending round the suction pipe. During the state from A to B, therefore, the local vortex appears to have no effect. This plus the decrease in effect of column vortex and the unstable conditions with increase in submergence make the discharge increase from A to B. B is the critical point at which the effect of local vortex comes into play.

Between B and C (Fig. 3), with increasing submergence, the effect of local vortex becomes more pronunced, mthe distortion of the local vortex, in bending round into the suction pipe is reduced and more and more air gets entrained (Fig. 4). Hence the discharge decreases with increase of submergence from B to C.

The effect of local vortex is most predominant at C (Fig. 3) and any further increase in submergence makes it unstable. Due to this instability of the vortex, air core will enter the suction intermittently. The requency of the vortex just forming and breaking will reduce with increasing submergence. During this period the eddies in the region of suction columns become slow-moving and the tendency for vortex formation is also reduced. This state from C to D (Fig. 3) may be treated as an unstable state for the formation of a local vortex and it may be noted that with increasing submergence, discharge increases from C to D. Beyond D, there is adequate submergence and hence there is no effect of any vortices. The curve follows the normal head- discharge relationship beyond D.

(*ii*) Conditions for the formation of Vortices:—

In order to study the various conditions under which different vortices would form in a sump, a glass tank of 3 ft. × 2 ft. × 4 ft. deep provided with a beffle and stilling arrangement was used. The suction pipe was so located in the chamber as to be free from wall effects. A closed circuit was laid out with the delivery pipe leading back into the tank (Fig. 5). Fine coloured threads tied at various points on a specially prepared wire grid work was kept at different positions near the suction pipe, to get an idea of the flow pattern in the sump at varying values of submergence. A similar grid work was kept inside the transparent lucite pipe connections made at the suction bell-mouth and near the impeller eye.

Under full discharge conditions, more rotation of flow was noticed in the pipe just above the suction bell than near the impeller eye, showing thereby that all rotation starts lonly from the suction sump. It was also observed that water just above the suction bell on the upstream side flowed straight into the mouth whereas water near the surface flowed past the inlet pipe and entered the suction bell from the downstream side.

A 'Column Vortex' may be formed:

(*a*) due to a greater swirl just around the suction column the vortex becoming concentric with the suction pipe and the draw down around the periphery allowing for air-entrainment; and

(*b*) due to the creation of a free vortex by the inflow into the suction pipe similar to those formed while emptying a wash basin or as in the Volute siphon.

A 'Local Vortex' is the result of eddies in the sump. Any boundary configuration of the sump favourable for eddy formation is a sure source of vortex. Sharp corners, angular piers, etc., result in irregular velocity distribution and hence give rise to local vortices. One unavoidable source of disturbance in the sump is the suction pipe itself.

Flow past the suction pipe is one of the reasons for eddy formation due to wake effect. This means that for a uniform flow in a symmetrical sump, the vortex is formed on the downstream side of the suction pipe. Sometimes the flow comes back and again flow past the suction pipe causing eddy formation on the upstream side. It was interesting to note that more air was entrained by a vortex formed on the downstream side than by the one formed on the upstream side. This incidentally explains why there is a clear reduction in vortex formation when the clearance between the suction bell and the end wall is restricted.

The vortex formation may be due to an appreciable swirl in the sump. In order that this eddy may give rise to a local vortex, it should have enough strength to make a deep depression. Be sides this, a strong downward flow towards the suction inlet adjacent to the eddy is necessary before an air core is set up.

A vortex, is thus formed by a combination of a swirl and a sink and its strength depends upon the relative strength of the two. Even though an appreciable swirl might exist in the pump, no vortices may form at all unless there is a downward flow towards the suction inlet. In a like manner just the existence of a downward flow without any swirl does not cause vortex formation. This can be easily demonstrated by pumping out water from a sump without returning the delivery back into the sump.

(*iii*) Effect of Sump Geometry on Safe Submergence:—

From practical considerations, the above discussion on the phenomenon of vortex formation can be analysed based on the following factors:

(*a*) *Shape of the sump*:—This is a factor of much importance which decides the flow pattern in the sump and hence the whirl conditions.

(*b*) *Size of the sump*:—This is of importance from the point of view of economy and the clearances that can be allowed.

(*c*) *Size of the Suction pipe*:—In the open sump, the eddies and their strength depend upon the wake effect resulting from the flow past the suction pipe which in turn depends upon the diameter of the suction pipe.

(*d*) *Location of the Suction pipe*:—The location of the suction pipe decides the clearances from end wall, the side wall and the bottom, which the affect vortex formation.

(*iv*) **Effect of Suction Flow Characteristics on safe submergence:**

(*a*) *Inlet Velocity*:—The strength of the vortices depends upon the strength of the downward flow towards the suction inlet in the neighbourhood of the eddy and this downward flow, in turn, is dependent upon the inlet velocity. In the case of local vortex, a steep rise can be seen in the value of safe submergence as the velocity changes from 4 ft. sec. to 6 ft. sec. (Fig. 6(a) & (b). Inlet velocities less than 3 ft. sec. have no effect on local vortices.

(*b*) *Velocity of inflow into the Sump*:—The strength of the swirl in the sump and that of the eddies due to the flow past the suction pipe, depends upon the velocity of inflow.

(*c*) *The type of inflow into the sump*:—Vortices may some times be destroyed due to disturbed condition of inflow into the sump. The safe submergence for different types of inflow conditions appears to follow no definite law and its effect can best be determined by model studies. Although scale models cannot give accurate quantitative results, because scale effects are not properly understood at present, they do indicate ways of improving the designs and possible remedies should air-entraining vortices occur in practice.

(*v*) **Suggested methods to prevent vortex formation**:—

Some of the earlier suggestions by other investigators have already been noted under the review of existing literature. Vortex formation can be prevented by restricted clearance between the suction pipe and the end wall as well as the clearance between the suction mouth and the floor of the sump. This, of course, is dependent upon the pump performance, which forms the limiting condition. Iversen noted that the side wall and end wall clearances in the range of D/2 to D/4 (D=Dia. of the suction pipe) and a bottom clearance of D/2 did not affect the pump performance, But even undert hese conditions Vortex formationan be seen with bad inflow conditions.

Vortex formation can also be prevented by providing suitable guide vanes in the suction bay to straighten out the flow into the suction pipe. A sump made out of G. I. sheet similar to the one shown in Fig. 7 was tried at the Indian Institute of Science, with success. It prevented Vortex formation to a great extent even under low submergences.

A conical cap with flattened fins (Fig. 8) also made of G. I. sheet to break up the vortices, was also tried with success. This cap can be attached to the suction pipe at the water surface. By a float arrangement its position can be automatically adjusted for varying submergences. Table II shows the values of discharge for varous submergences with and without the finned cap. The Graph (Fig. 9) clearly shows that there is a reduction in the effect of both types of vortices by the finned cap. As this fixture is very simple and inexpensive to make it should prove to be of great benefit to the pump users.

Table II

Sl. No.	*Submergence in ft.*	*Discharge in Cusecs.*		
		Without cap.	*With conical cap (No flattened fins*	*Conical cap with flattened fins.*
1	0.22	0.3030	0.3510	0.3627
2	0.25	0.3067	0.4267	0.4415
3	0.30	0.4375	0.5412	0.5840
4	0.35	0.5837	0.5867	0.5972
5	0.40	0.5887	0.5892	0.5972
6	0.50	0.5783	0.5867	0.6000
7	0.55	0.5709	0.5840	0.6000
8	0.60	0.5660	0.5840	0.6000
9	0.65	0.5672	0.5840	0.6000
10	0.70	0.5790	0.5867	0.6025
11	0.75	0.5867	0.5902	0.6025
12	0.80	0.5892	0.5945	0.6025
13	0.90	0.5902	0.5945	0.6050
14	1.00	0.5945	0.5945	0.6070
15	1.05	0.5972	0.5972	0.6070

CONCLUSIONS

From the studies made by several investigators at the Indian Institute of Science, it is clear that the and factors governing the formation of air-entraining tices are so numerous tht no de finite generalisatiCn in the shape of a formula is possible. Model studies of any particular problem concerning intake studies of a hydraulic machine would be very great utility and some times indispensable in order to design vortex-free suction walls, having due regard to the shape of the sump, submergence, condition of the sump (open or closed), shape of the suction bell, location and size of the suction pipe, etc.

The devices suggested in this paper, viz., a suitable sump geometry (Fig. 7) and a flat finned conical fixture to the suction inlet (Fig. 8) could be used with advantage as they are inexpensive, simple and effective.

It is recommended that the suggestions given in this paper may be used as a guide in designirg sumps fo pump installations.

[*Continued on page 76*

Similarity under Cavitation and Scale effects in Model Testing of Hydraulic Machines

By B. P. SINGHAL

Cavitation has been classified under two major headings for scaling effects:

(1) *Appearance of first vapour bubbles:* With prototype velocities, the life of the bubbles for expansion and re-sorption is much shorter than in reality and this suggests an examination of the influence of gradual velocity reduction and longitudinal pressure gradients to determine whether there is a lower thresh hold transit time for the scale model.

(2) *Developed cavitation in different forms accompaying progressive pressure reduction*: (*i*) Intense bubble emission in the mass of moving water. If the transit time is sufficient, the bubbles collapse where the local pressure is equal to the critical pressure. Froude similitude should disclose the bubble path between formation and collapse but may introduce distortion of the bubble dimension from which it is concluded that it is desirable to use (*a*) prototype velocities for bubble dimension analysis, (*b*) Froude similitude for definition of the bubble formation and collapse zones, (*ii*) Cavitation attached to the surface. This form is initially unstable and subject to rapid cyclic formation and detachment needing prototype velocities for simulation.

(*iii*) Established cavitation where the bubbles become larger and are partially filled with vapour. The cavity dimensions are related to the surface pressure gradients and consequently the variable hydrostatic pressures should be correctly represented on the model by Froude similitude specially if the cavity extends into the downstream zones.

Scale effect: By scale effect is meant the effect of the size of turbine on its efficiency in a series of homologous turbines.

(*i*) The so called homologous turbines of different actual dimensions, geometrically similar in design and proportions, are not completely homologous with respect to surface roughness.

(*ii*) In efficiency tests of turbines, there is a wide range of possible error both in the field and laboratory due to the difficulty in measuring the water quantity, field measurements of efficiency, similarity in turbulence, etc.

Model Tests are based mostly on considerations of Thoma Number. Geometrically similar models representing the entire turbine or pump are necessary. Tests should be done under velocities similar to that in the proto type.

Inception of cavitation may obey similarity laws while extent of cavitation damage may not be scalar under conditions when inception of cavitation can be reproduced under homologous condition (K_i number being the same in model and prototype).

GENERAL PROBLEM OF SIMILARITY

The problem of similarity of flow conditions may best be described by the question, under what conditions will a geometrically similar flow of liquid or gas occur around or within geometrically similar bodies? The general answer to this question is obviously that not only the guiding surfaces but also the forces acting on the fluid must be similarly arranged, because otherwise the fluid would be forced off the geometrically similar path inspite of the similarity of the fluid passages.

This problem can be treated under the simplest possible assumptions. One of these assumptions is that the inertia forces are the only forces acting in the fluid. Furthermore, the fluid is considered as incompressible and, generally, as incapable of hanging its physical properties while passing through the machine.

The similarity considerations carried out under these assumptions are called "fundamental" in contrast to those concerning viscosity, compressibility, and vaporisation (cavitation) which require other considerations.

The "fundamental" similarity relations are:

$$\frac{V}{U} = \text{constant (Kinematic similarity) and}$$

$$\frac{2gH}{V^2} = \text{constant.} \quad \frac{2gH}{U^2} \text{ constant}$$

(Dynamic similarity)

Where V = fluid velocity
U = peripheral velocity
H = total head on the machine

The geometric similarity leads to the requirement.

N_s = constant
N_s = specific speed.

SCALE EFFECTS AND MODEL LAWS

(*a*) *Scale effects* :—Similarity considerations are concerned with changes in the absolute dimensions

and/or velocities of the system that do not effect the geometric and kinematic similarity. Such changes may therefore be described as changes in the "Scale" of the dimensions and/or the velocities of the system. Under the influence of such scale changes, those effects which cause departures from fundamental similarity relations are called "scale effects."

Thus "Scale effect" is the effect of the "size" of a hydrotechnical device in a series of homologous devices.

(*b*) *Laws governing models:* The dimensionless groups most commonly used in hydraulic experimentation are designated as Froude's number, Reynolds' number, and Weber number.

(1) Froude's number

$$F = \frac{Vv^2}{Lg}$$

expresses the condition for similarity of gravity and inertia forces.

This simple application supposes that neither viscosity, nor surface tension have any appreciable influence on the phenomenon under investigation. Froude's number is used as the basis of design and interpretation of models in which friction forces are negligible, such as spillways and other structures in which there is rapid change in the elevation of water surface in short distance.

Froude number similitude generally involves free surfaces, certainly cavitation has a free surface, and gravity forces are sometimes important in the behaviour of cavitation regions. Air-water entry of missiles is an example.

(2) Reynolds number.

$$Re = \frac{VL}{\nu}$$

states the condition under which the ratios of viscous forces to inertia forces are the same in models and prototypes. Gravity and surface tension are neglected in the use of this criterion.

Reynolds number is chiefly applicable to closed systems of flow, such as pipe or conduits, where there is no free water surface. Cavitation criterion in such systems is associated with Reynolos number.

(3) Weber number.

$$We = \frac{V^2 L \rho}{\sigma} \quad \sigma = \text{surface tension}$$

expresses equality of the ratios of the surface tension to inertia forces in model and prototype and is useful in studies of surface waves, formation of drops and bubbles, entrainment of air in flowing water and other related phenomenon. The use of Weber number as a criterion supposes that forces of gravity and viscosity are negligible.

The inclusion of Weber number as a cavitation criterion is certainly justified.

CAVITATION INDICES

(*a*) *Cavitation parameter:* The cavitation parameter is a dimensionless ratio. It is

$$K = \frac{P_o - P_v}{\frac{1}{2}\rho V_o^2}$$

where V_o = Velocity of the undisturbed flow

P_o = Pressure in the undisturbed flow

P_v = Vapour pressure at the liquid temperature.

Basically, the cavitation parameter defines the conditions of the flow in a given cross-section. It can be used in two different ways. The first is to describe the condition of the flow in a given cross-section without regard to whether or not cavitation is being produced. The second refers to a guiding surface confuguration. Here the use is to discribe the flow conditions over this guiding surface corresponding to a certain degree of cavitation. The value of the cavitation parameter for which the cavitation just appears in a non-cavitating flow or just disappears in a cavitating flow on a given surface is known as K_i, the subscript, *i*, indicating the inception point. If the flow conditions are changed so as to reduce K below K_i, each new value of K will correspond to a given degree of cavitation, i.e., a given length of the cavitation zone. This implies that in a cavitating flow, geometric similarity requires that the value of must be the same for model and prototype.

(*b*) **Thoma Sigma:** In simple flow systems there is usually little difficulty either in measuring or calculating the value of Katany given point in a system. However, in hydraulic machinery this is not the case. For such systems Thoma number, first published in 1925, is used. The relation is not limited in application to machines having moving parts, but is equally applicable to stationary conduits (open system or closed system). Its definition in the usual presentations is.

$$\sigma = \frac{H_a - H_{vp} \mp H_s}{H}$$

$$= \frac{H_{sv}}{H}$$

where H_a = atmospheric pressure head

H_{vp} = Vapour pressure head

H_s = Static suction head,

It can be both positive or negative

H = Total head on the machine

H_{sv} = Net positive suction head, o: NPSH.

Thoma maintained that if similarity of flow was to exist under cavitation conditions, the extent of the cavitation zone, or vapour pocket, relative to the fluid passages must remain unchanged. The fundamental dynamic relations apply not only to the total head H of the machine, but also to any other pressure or head difference in the machine. The existence of the vapour pressure in the zone of cavitation can then be taken into account simply by the requirement that the difference between this pressure and any other pressure in the fluid surrounding the vapour pockets must follow the dynamic similarity relations. Since H_{sv} represents the difference between the

total head at the point where suction head is measured and the static pressure in the region of cavitation (Vapour pressure), it follows that similarity of flow will be maintained if this head difference H_{sv} follows the same dynamic relations as the total head H on the machine, or any other head difference in the machine. Thus making H_{sv} proportional to H gives the Thoma law.

Thoma also pointed out that "a rigorous mechanical similarity between model and prototype is only possible if the ratio H/D is the same for both". D is any representative dimension. This simply means that Thoma stipulated compliance with Froude's law.

(*c*) Suction Specific Speed.

$$S = \frac{n\sqrt{Q}}{H_{sv}^{3/4}}$$

The suction specific speed may be defined as that combination of inlet operating conditions which, if held constant, permits similar flow and cavitation conditions in geometrically similar passages of centrifugal and axial flow pumps. Its value characterizes the hydro-dynamic design of the inlet passages with respect to local pressure reduction that can lead to cavitation.

S is related to σ by

$$\sigma = \left(\frac{n_s}{S}\right)^{3/4}$$

where n_s is the specific speed.

The Hydraulic Institute limit for mixed flow and propeller pumps

($n_s > 4{,}000$) is

$$S = \frac{rpm\sqrt{gpm}}{H_{sv}^{3/4}} \leq 8140$$

or

$$S = \frac{rpm\sqrt{cfs}}{H_{sv}^{3/4}} < 384$$

PHYSICAL LIMITATION OF SIMILARITY CONSIDERATIONS ON CAVITATION

Similarity relations do not include the effects of Viscosity and compressibility of the fluid involved. Since cavitation involves the formation of surfaces separating the liquid from the vapour phase of the fluid, surface tension of the fluid need also be taken into account.

For viscous fluids, it is necessary to determine the cavitation characteristics of a machine experimentally for different Reynolds numbers. However, although it is usually possible to make some qualitative predictions regarding the influence of viscosity on other hydraulic characteristics, particularly the power and efficiency of the machine, no general information is available to indicate even the direction in which an increase in viscosity (large reduction in Reynolds number) will influence the cavitation characteristics of the machine. This problem is complicated by the fact that highly viscous fluids do not necessarily have one definite vapour pressure.

Another important factor is the "time" during which a fluid particle remains in the low pressure zone. Under the influence of surface tension, the rate of increase of bubble size is not constant, and in the large machine it may not be similar to the size in the model. Therefore, if the time required for the growth of the bubbles under the influence of surface tension is appreciable, similarity in the size of the vapour bubbles in the large machine and in the model cannot be expected.

The other physical property of the fluid not taken into account is compressibility.

Since cavitation has significance only in connection with liquids, i.e., fluids that are not normally considered compressible, it would seem that neglecting the compressibility does not lead to appreciable errors. This is true of all hydraulic phenomenon with the exception of collapse of vapour pockets. Here the pressures are very high, and neglect of compressibility may lead to errors.

Cavitation effect is very similar to "water hammer", in which case it is not possible to neglect compressibility. The pressures caused under this condition are proportional to the product of the fluid velocities and the velocity of the pressure waves in the media considered. The consideration of compressibility, therefore, leads to the result that for a given fluid with a fixed accoustic velocity, the pressure of cavitation collapse increase according to the first power of fluid velocities, in contrast to other pressures in the machine which very as the square of the velocity.

CAVITATION SCALE EFFECT

(*a*) Scale effect at inception of cavitation:—The most difficult modelling problem is that of scaling cavitation onset. Cavitation onset is governed by three parameters. (1) the characteristics of the impurities in the liquid which form the nuclei, (2) the physical and thermodynamical properties of the fluid, (3) the hydrodynamic characteristics of the flow.

The interaction of these three parameters results in a time delay in the inception of cavitation, which introduces a scale effect, and a brief analysis of the problem leads to the following tentative conclusions:

(*i*) The same body tested at different velocities will have different values of K_i;

(*ii*) If similar bodies of different sizes are tested at the same velocity, the smaller body will have lower K_i;

(*iii*) Similar bodies of different sizes tested at the same value of the product (Velocity × characteristic diameter or length) should have the same K_i.

Of particular interest here are the results reported by Kermeen and Parkin and Holl. In experiments on bodies of revolution having diameters ranging from $\frac{1}{4}''$ to $2''$, they observed a definite dependence of the critical cavitation number K_i, on model size and absolute flow velocity.

The dependence of inception on total air content has been reported by Crump which is in variance of

observations of Kermeen who found no consistent effect of air content varied between 7 and 13 ppm. Also, incipient cavitation number, defined as the state of liquid flow at which cavitation just disappeared, varied with free stream velocity and model scale. Lerbs reported a critical Reynolds number for inception, in contradiction to the measurements of Kermeen.

(*b*) Scale effect at partially and fully developed cavitation :—

There is some qualitative evidence to show that the scale effect is less for partially and fully developed cavitation than for inception. And it is possible to reproduce characteristic changes when cavitation is sufficiently developed. Although designers endeavour to prevent the appearance of cavitation, it must be tolerated under certain transient conditions runaway and partial load operation.

(*c*) Scale effect in hydro-dynamic performance:— Cavitation affects performance by altering the shape of the water passages, and any difference in geometric similarity in the cavitation Zone between model and prototype will produce a scale effect which will become less as the physical size and test velocity of the model are increased. This seems to point to the desirability of cavitation testing at prototype Reynolds number. In the case of low Froude numbers gravity forces exert a significant influence on cavitation behaviour and in such cases scaling should be at prototype Froude number. Thus there is a scaling dilemma, to obtain correct cavity geometry Froude number scaling is indicated, but this involves low velocities and small dimensions, in which conditions cavitation inception is delayed and cavitation zones are smaller than they should. Fortunately this dilemma becomes serious only in the case of low Froude numbers and testing is usually carried out at ashigh a velocity as possible.

The scale relationships for equipment operating in the incipient and fully developed cavitation zones in terms of Frounde's number call for examination because of the many variables and physical phenomen on involved. It seems that because of different characteristics of cavitation one law is insufficient and consequently each phase must be analysed and appropriate sealing laws applied in association with prototype values of the Thoma cavitation parameter. Difficulties, however, arise in reproducing zonal similarity where incipient or limited cavitation tests require model velocities approaching or exceeding those in prototype.

Applying these considerations to hydraulic machinery, it is clear that for low head turbines disregard of the elevations of different parts of turbine can introduce errors in determining turbine setting much in excess of those applicable to model test measurements.

(*d*) Scale effects in cavitation induced vibrations, noise and erosion.—

Due to vibration, noise and erosion which accompany cavitation difficulties arise in reproducing such phenomenon on scale models. Since (*i*) Vibration and noise introduce the physical properties of water, hydraulic phenomena and mechanical construction of the turbine apart from prototype and structural similitude which obey different incompatible laws. The best that can be done is to observe periodic pressure fluctuations at different points in the machine. (*ii*) The laws of erosion similitude even if determined for simple shapes with prototype velocities could not necessarily be extended to turbine model. Reproduction of erosion would take too long and interpretation would be too difficult in the present state of metallurgical knowledge. A large model at prototype Froude number, however, could predict the zones exposed to erosion where highly resistant materials should be used. When coupled with a study of vapour bubble behaviour, it might indicate where improvement is desirable, and perhaps define intensity but not the severity of erosion damage.

It is a known fact that pitting occurs not in frequently in Zones which on the model were cavitation- free or, conversely, that no damage occurs in sections which from model observations were considered endangered by cavitation attack. Hydro-dynamic similarity with regard to cavitation therefore by no means signifies similarity of cavitation damage.

CAVITATION MODEL TESTS

Th complexity of the cavitation problem rules out theoretical solutions. Model tests have become the indispensible tools of research and engineering development work.

From dimensionless equations of motion, it follows that Reynolds number, Froude number and the cavitation parameter must be the same for model and full- scale machine. If Weber number is also included, then

K or $\sigma = f$ (Re, F, We, test geometry)

where test geometry extends to roughness and clearance parameters.

Reynolds number effects are assumed insignificant if the tests are carried out above the critical range of transition, for axial flow machine, this is, in general, assumed as Re 2 to 4×10^5, if based on blade chord.

Froude number has always been considered an essential parameter since gravity forces are the only external forces acting. This leads to:

σ = constant; F = constant

The tendency today favours cavitation tests on complete scale models, a practice frequently adopted in model efficiency tests.

Cavitation test procedure for model turbines consists of determining unit power and discharge as a function of the cavitation parameter σ. The critical conditions are determined from the break in either the unit power or discharge curve, whichever occurs at higher value of σ.

For centrifugal pumps σ crit is found from the break in head-capacity curve.

Coating techniques are some times used to locate the parts likely to be attacked by cavitation.

Cavitation model tests on ship propellers are conducted. with the cavitation number, referred to the

centre of the axis, as the predominant parameter and with the Reynolds number chosen greater than a certain critical number.

Cavitation model testing is expensive which is one of the reasons why it has increasingly been supplemented by aerodynamic methods which also offer far greater flexibility. Flow in-component investigations using air and the development of suitable blading for high solidity runners of modern axial flow machines in aerodynamic cascade tests is today industrial routine procedure. This has been extended to complete air models of hydraulic turbomachines and even to the determination in such tests of the critical cavitation coefficient.

Aerodynamic investigations, however, are limited in so far as they cannot supply information on performance under conditions of cavitation advanced from the incipient stage, nor can they be used for reliable prediction of onset of separation or vortex cavitation The critical cavitation coefficient of the air model tests will therefore be an inadequate criterion in all those cases where plant operation dictates acceptance of tolerable cavitation in the machine, or where an indication is wanted of the possible danger of pitting due to gap or clearance cavitation.

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Some Studies on Air-Entraining Vortices in Pump Sumps

It can be easily seen that there is plenty of scope for further work in this field.

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Remarks on Cavitation in Turbomachines

By A. J. ACOSTA and A. HOLLANDER

The occurrence of vapour and gas filled cavities within the body of a flowing liquid have been a major source of concern to the user and designer of turbomachines practically from the time these devices have come into being. This phenomenon, called "cavitation", can result whenever the static presure of the flowing liquid falls below the vapor pressure of the liquid or in the case of a liquid saturated dissolved gas, the bubble point. The low pressure regions of a cavitating fluid may appear to be frothy as when gas bubbles come out of solution, may consist of a series of growing and collapsing bubbles filled predominantly with vapor, or even in some cases a fairly distinct cavity attached to a solid surface may form. In all cases the hydrodynamics of the flow field can be affected to a marked degree by the presence of such cavities, and equally important, the pressure waves originated by the collapse of transient cavities in the flow may result in severe material erosion known as cavitation damage. Various types and extent of cavitation are shown in Fig.1 as they occur on a body of revolution.

These two general areas, i.e. hydrodynamic performance and damage, serve as general focal points for most of the cavitation research insofar as its connection with machines is concerned, that has been carried out in the United States and Europe. The literature of this subject began essentially with Rayleigh[1] who mathematically formulated the pressure rise due to the collapse of a spherical bubble, to explain the origin of cavitation damage. Since that time many papers have appeared that discuss physical and metallurgical aspects of cavitation damage. A comprehensive summary of these and related problems is given by Knapp ([2]). The hydrodynamic performance of some simplified shapes in cavitating flow can be determined with the aid of free streamline theory when the cavitating region is fairly distinct and well defined. The textbook example of such a flow is, of course, the plane flow past a lamina held normal to the stream, the pressure in the cavity being equal to that far upstream. This early example due to Kirchoff predicts a drag coefficient that agrees well with experiment[3]. Many more examples are found in the book by Birkhoff and Zarantonello[4].

Unfortunately, not all problems of technical importance satisfy the neat conditions of two-dimensional, steady, potential flow that must be satisfied in order to use the powerful method of analysis outlined in ([4]). As a result, most investigations of cavitation in such complicated machines as turbines, centrifugal pumps, marine propellers etc. are perforce, experimental. The object of such researches is to establish the safe limits of operation with cavitation not only for performance but for material damage as well. As a rule a certain arbitrary reduction in efficiency such as one-half or one percent is adopted as adequate insurance against cavitation damage. The acceptable deterioration in performance is, however, dependent upon the length of anticipated service (and thus extent of damage) and upon the particular application. Thus, for example, the extensive cavitation permitted in a modern missile propellant pump would not be tolerated in the pumping plant of a major irrigation project. It is, however, always necessary to be able to establish safe hydrodynamic limits of operation with cavitation and for this reason interest in the present paper will be centered on the hydrodynamic aspects of cavitation rather than on material damage.

In the following sections several aspects of the development of cavitation in a turbomachine will be taken up. Some of the current theories of design for cavitating flow will be discussed and finally, some aspects of the important problem of similarity in a cavitating flow will be mentioned.

DEVELOPMENT OF CAVITATION IN A MACHINE

The lack of detailed flow observations in machines operating with various degrees of cavitation has hampered real understanding of this problem. Visual observations of cavitating flow in a centrifugal pump, for example, are practically non-existent in the literature. One of the best papers dealing with overall performance measurements and their interpretation is, however, due to Gongwer[5]. The situation is scarcely better with axial flow pumps. The early paper of Tenot[6] showing photographs of cavitation development in an axial flow pump is interesting. A criticism of Tenot's work, together with additional photographs of cavitation development in an axial pump, is given in a less available report ([7]). A more detailed and comprehensive photographic survey of cavitation phenomena in an axial inducer pump (a pump characterized by extreme solidity to achieve great resistance to cavitation failure at the possible expense of efficiency) is given in ([8]) again, unfortunately, not a part of the standard literature. The sequence of events as found in these works bear several features in common and it is anticipated that the general features of cavitation in a radial machine will probably not be greatly different from that in an axial machine. However, absence of photographs of cavitation in a centrifugal impeller precludes any definite statement.

In general it is found[7,8] that as the inlet pressure is reduced, the progress of cavitation undergoes several

recognizable stages enumerated below for an axial pump at a constant flow rate and rotative speed:

(*a*) Inception: The first appearance of cavitation is always in the region of lowest pressure. If the flow is separated or if there is a strong tip clearance flow, cavitation may commence in the stream usually appearing in a tip vortex which may or may not be attached to the blade. Figure (2) shows such a case. When the tip vortex does not form, cavitation then appears first on the blade surface at or near the position of minimum pressure.[9]

(*b*) Partial cavitation: With continued reduction in inlet pressure the cavity grows. The tip vortex region, if it occurs, becomes attached to the blade and the region of surface cavitation gradually extends over the chord of the blade. Generally speaking when the length of the cavity is less than about one-half chord, no gross effect on head generation is observed provided the cavitation commences from the leading edge of the blade. Still, however, the head may be lowered by several percent and the efficiency perhaps even more. If the impeller solidity is large (say 1.5) a condition of non-steady flow may also develop, depending upon the particular blade design. Also, observation of partial cavitation on Karman-Trefftz airfoils when the cavity is about on-half chord long[10] indicate that pumps of low solidity will probably be subject to similar non-steady cavitation flow. The exact origin of the unsteadiness is yet unknown. It is, however, a phenomenon of some importance as pumps that operate with such flows are subject to cosiderable vibration.

(*c*) Impending Failure or Cavitation Breakdown: With further decrease in inlet pressure, the length of the cavitating region (steady or unsteady) rapidly approaches the chord length and the head of the unit may suddenly drop to a very low value. For practical purposes the pump may be said to have failed. Examples of such behaviour are common in the literature and the representative plots in [11] may be cited as being typical. It is just this condition that the designer must be able to estimate in order to determine the degree of safety of a particular application. Moreover, there are circumstances such as in the regulation of condensate pumps in which it is desirable to operate with various heads within the region of "cut off"

The foregoing developments of cavitation in an inducer impeller are illustrated in Fig. 3 taken from ref. ([8]). The gradual development of the cavity is in this case regular and no unsteady flow occurs. It will be noticed, however, that extremely low cavitation numbers (see notation) are achieved before breakdown. It will be seen later that this is a consequence of the low blade angle and high solidity of this configuration. The dimensionless head coefficients are plotted vs the cavitation numbers for various flow rates (Fig. 4) for this impeller. It is clear from this diagram that to a certain extent, the acceptable safe limit is arbitrary, provided that it exceeds the break-down limit. It becomes therefore imperative to have some measure of the breakdown limit as well as an understanding of the physical processes implicit in the use of the dimensionless parameters such as the cavitation number used in this diagram. The first of these problems is the next to be taken up.

ESTIMATES OF CAVITATION PERFORMANCE

1. **Similarity parameters for cavitation:** The cavitating performance of a hydraulic machine in the field is usually expressed in terms of suction lift at a given flow rate and speed. However, such a presentation is not useful for application to other pump size, speeds or lifts and for that reason, representation in terms of dimensionless similarity parameters is always preferred. Various parameters are now in vogue among hydraulic engineers for this purpose. A convenient one (and one of the first) is Thoma's σ

where

$$\sigma = h_{sv}/h \qquad (1)$$

In this equation h_{sv} stands for the net positive suction head and this is equal to the inlet total head minus the vapor pressure of the fluid. h is the non-cavitating head of the pump. At a given flow rate coefficient (see notation) it can be shown that constancy of σ implies constancy of the cavitation condition provided other physical effects do not occur (such as important Reynolds number effects, non-equilibrium thermodynamic effects, surface tension effects etc.) However, since the magnitude of the generated head affects only to a slight degree the distribution of pressure in an impeller of moderate solidity, the magnitude of σ is not a sensitive indicator of cavitation performance and varies greatly with impeller type. For this reason the cavitation number

$$K = \frac{(p_1 - p_v)}{\frac{1}{2}\rho w_1^2} \qquad (2)$$

or the suction specific speed

$$S = N\sqrt{Q}/(h_{sv})^{3/4} \qquad (3)$$

are better parameters. In Eq. (2), p_1 is the inlet static pressure to the impeller, p_v the vapour pressure, the density and u_1 is the inlet relative velocity at the tip of the impeller. The algebraic relation between (1), (2) and (3) is brought out more directly by introduction of the dimensionless head coefficient and flow coefficient (see notation). Making the indicated substitutions it is found that

$$K = \frac{2\Psi_2\sigma\left(\frac{r_z}{r_1}\right)^2 - \Phi_1^2}{1+\Phi_1^2} \qquad (4)$$

For axial flow machines $r_z = r_1$ so this ratio disappears. The suction specific speed is

$$S = \frac{8150\Phi^{1/2}_1 r_z}{[(1+\Phi_1^2)K+\Phi_1^2]^{3/4}\, r_z\sqrt{A_1/\pi r_1^2}} \qquad (5)$$

a form suitable for either centrifugal or axial pumps. For the latter, the last square root number becomes $\sqrt{1-\nu^2}$. In Eq. (3), N is to be measured in RPM, and Q in gallons per minute (US). S, as found is not dimension less but would be so with the introduction of $g^{3/4}$ into the denominator ($g=32.2$ ft/sec,^{2} the standard gravitational constant).

From these relations it is seen that if the operating point of a given turbomachine is fixed (Φ, Ψ const.) constancy of σ implies constancy of k or S.

2. **Estimates of Cavitation Parameters:** The problem now is simply to be able to estimate values of k or S such as to ensure freedom from cavitation breakdown. This is, regretably, a most difficult problem. What can be done for simplified shapes (for example, cascade of hydrofoils) is to estimate the inception of cavitation by equating the minimum pressure on the blade to the vapour pressure of the fluid. It is particularly simple to do this for pump blades of low solidity and for this case Bowerman[12]. has devised a neat procedure for designing pumps of given duty (head, flow, speed) for given diameters with optimum resistance to (the inception of) cavitation.

However, if it is known that a certain type of pump will operate successfully with a particular value of k, independent of Φ or Ψ, a special case of Bower man's theory results which enables one to determine for example the optimum diameter of the impeller eye easily. Such considerations were first put forward by Brunfield[13] Pfleiderer[14] and Ross[15]. In fact due to its simiplicity this calculation (by Ross is worth repeating:

Let the inlet total pressure be p_o. Then in the notation of Fig. 5,

$$p_1 = p_o - \frac{\rho}{2} C_m^2$$

and the minimum pressure on the blade, assumed to be to equal to p_v is from (2)

$$p_v = p_1 - \rho/_2 \, w_1^2 = p_1 - \rho/_2 \, (v_1^2 + C_m^2) \, k$$

or

$$p_v = p_o - \rho/_2 \, [k v_1^2 + C_m^2 \, (1+k)]$$

The optimum diameter of the impeller eye for a given speed and flow rate will occur when p_v has its highest value or when $k_1 V^2 + C_m^2 \, (1+k)$ has its minimum value. Thus

$$0 = \frac{d}{dr} \left[k \, r^2 w^2 + (1+k) \frac{Q^2}{\pi^2 r^4} \right]$$

is the condition to be satisfied for the optimum radius. Carrying out the indicated operations results in relation.

$$\gamma_{1\,opt} = \left[\frac{2(1+k)Q^2}{k \pi \omega^2} \right]^{1/6} \qquad (6)$$

A more interesting result is obtained when the optimum flow rate coefficient is found

$$\Phi_{1\,opt} = \sqrt{\frac{k}{2\,(1+k)}} \qquad (7)$$

and with this

$$S_{1opt} = \frac{5060}{k^{1/2}(1+k)^{1/4}} \sqrt{A_1/\pi r_1^2} \qquad (8)$$

Although (8) is for the optimum eye diameter, it will be found that (6) also maximizes (5) for the given conditions. Thus, the radius given by (6) is the best radius, the flow coefficient of (7) and the suction specific speed of (8) are the best that can be achieved with the acceptable value of k. It is interesting to determine the order of magnitude of these quantities for some realistic estimates of k. It can be seen upon reference to Fig 4 that $k=0.03$ is certainly achievable for some conditions. For this value of k the optimum flow coefficient is from Eq. (7), $\Phi_{1\,opt}=0.12$ and the value of S_{opt} is 26,800., (it is assumed that $r=0.4$). This value is quite high and would only be used in machines of short life.

These numbers are in interesting contrast to more conventional pumps. For example, the data of [16] indicate that a value of k of about 0.31 can be expected near breakdown for a well-designed centrifugal pump. Under this condition, the best suction specific speed that could be expected would only amount to 8500. This value is in close agreement with the Hydraulic Institute standard for single suction centrifugal pumps.[17]

The development sketched above can be modified easily to include upstream friction loss, and pre-rotation of the flow. The advantages of pre-whirl in achieving high suction specific speeds (and therefore good cavitation resistance) is clearly brought out in a recent paper of Wislicenus [18]. However, in all of these cases *a priori* knowledge of the acceptable cavitation number is required. As mentioned earlier, this can be obtained analytically for inception of cavitation only in the case of some simplified flows such as a cascade. Even then considerable labor is required, for example, see [19]. The growth of the cavitating region in an impeller as the cavitation number is lowered and its correlation with performance is therefore a problem of great importance as this information would provide some basis for the calculations above. It should come as no surprise that the theoretical difficulties of treating the cavitating flow in a general turbomachine are practically insurmountable. For some simplified shapes, e.g. the impeller shown in Fig. 3, some progress has been made recently and this will be reviewed in the next few paragraphs.

As the simplest meaningful example that can be treated, the plane potential flow through a cascade of flat plates will be discussed in detail. The cavity is assumed to be attached to the leading edge of the blade (Fig. 6) and for reasons of simplicity, the length of the plates is taken to be infinite. The flow is frictionless, and the cavity is bounded by a free streamline along which fluid of constant velocity proceeds. The

position of the streamline and length of the cavity are unknown before hand. The problem to be solved is to determine the extent of the cavity (compared to the blade spacing 2π) as a function of the cavitation number k, the angle of attack α and the stagger angle γ. The solution can be obtained by hodograph methods which are abundantly treated in ref. (4). However, because of the recent interest in the application of linearized free streamline theory to problems that are handled only with considerable difficulty by exact methods, the linearized theory will be used herein.

Briefly, the method assumes that the angle of attack and the slope of the streamlines are small compared to unity (as in thin airfoil theory). As a consequence, it may be assumed that the basic flow through the cascade is only slightly perturbed from its value far upstream. Products and squares of the perturbation velocity components are then neglected, so that the presssure becomes a linear function of the perturbation velocity in the stream direction. Boundary conditions are then applied on the chord and not at the actual position of the streamline. The method approximate but because of its simplicity, it is of great utility. The linearized theory as applied to thin airfoils is covered in ref. (20), and Parkin (21) treats in detail a great many free stream-line problems by this method.

3. Discussion of Theoretical Results: The details of the problem outlined above may be found in Appendix I, and for the purposes of the present discusion the salient results will be repeated below. First, however, it is of interest to consider the results of two limiting cases; namely, the limit of zero cavity length to spacing ratio and infinite cavity length. It may be shown from Eqs. A-6 and A-11 that for small cavity lengths

$$k=\frac{4\,\mathrm{Sin}\,2\alpha}{\sqrt{2\pi}}\sqrt{\mathrm{Cos}\gamma/C/2\pi}$$

Thus, as the cavity becomes very short, k approaches infinity in accordance with the known limit for fully wetted flow. On the other hand, as the cavity length approaches infinity, Eq. (A-11) shows that k becomes asymptotic to the limiting value

$$k=2\sin\alpha.\cos(\gamma+\alpha)/(1+\sin\gamma) \qquad (9)$$

This result can also be obtained from momentum considerations when it is recalled that the flow is frictionless and that there must be no force parallel to the plane. Equation (9) is useful since it gives the least possible value of the cavitation number for flow through a cascade. It is seen that $k=0$ for $\alpha=0$ (it being assumed that there is no thickness to the blades), but in addition k is also zero for $\gamma+\alpha=\pi/2$. This finding, perhaps surprising at first, occurs only at zero flow through the cascade and does not therefore represent a flow of interest. The maximum of k in Eq. (9) is perhaps a more realistic condition and this occurs when $\alpha=(\pi/2-\gamma)/2$ and has the value $k_{max}=(1-\sin\gamma)/(1+\sin\gamma)$. This simple result clearly shows that large stagger angles are more conducive to cavitation resistance, and this observation is generally borne out in practice. In addition, there is some suggestion (5), (8) that the value of k_{max} just quoted is a fair measure of the minimum cavitation number achievable in an inducer pump or inlet portion of a centrifugal pump. Specifically, the test results of (8) show that twice the value of k so determined is a conservative estimate of the breakdown cavitation number in inducer pump devices.

The dependence of cavity length upon cavitation number for various values of γ and for an angle of attack of six degrees is shown in Fig. 8. The dominant effect of stagger angle is clearly brought out in this diagram, but equally interesting is the fact that the asymptote of minimum cavitation number is reached for all practical purposes when the cavity is one and a half blade spacings. The "knee" of the curve occurs at about one blade spacing for most stagger angles. This finding indicates that excessive solidity is not required to achieve high suction specific speeds and partial corroboration of this conclusion was observed in (8) for a limited range. Direct experimental evidence confirming the general behaviour shown in Fig. (8) is, however, lacking. To investigate this point visual observations of the cavitating flow in an inducer with a nine degree tip angle at an angle of attack for which three dimensional flows were not large, were made and compared to the theory. These results are shown in Fig. 9. The agreement, though not too good, certainly bears out the general features of the theory. It would therefore be of great interest to extend these observations to centrifugal pumps and to additional blade angles. The simple analysis indicated above can be extended fairly easily to account for finite length of the blades. An example of such a calculation for cavity lengths greater than the chord length is given in (22). Also, curved blades can be treated as well.[23] However, the effects of rotational flow on the development of cavitation have not yet been determined (i.e. that which occurs on the meridional streamlines in a centrifugal impeller). Much interesting and valuable work remains to be done in this area.

CAVITATION SIMILARITY

In the foregoing section it was tacitly assumed that the pressure in the cavity was equal to the vapor pressure of the bulk liquid. Furthermore, it was also assumed that cavitation occurs as a pocket or bubble attached to the blade surface. These assumptions now have to be examined in more detail, especially as recent experimental findings on pumps operating with hot water some hydrocarbons and cryogenic fluids such as liquid nitrogen, show pronounced differences in their cavitating performance. As an illustration, Fig. (10), taken from Stahl's paper [24] compares the performaance of the same pump operating with cold water and water at 294°C (the flow rate and speed are kept constant). It can be seen that the cavitating performance with hot water is appreciably better. In fact the results of numerous investigations 25, 26, 27, 28, show that all pure liquids containing no dissolved gas exhibit a cavitating performance equal to or better than that of cold water. One possibility that

immediately suggests itself is that the pressure p_v assumed to occur in the cavity, is lower than the vapor pressure of the bulk fluid, resulting in a higher cavitation number and therefore a less severe condition to be met by the pump. Various rough physical arguments can be put forward to support the view that this actually happens. For this purpose, it is necessary to have some idea of the processes involved in cavitation. One such model (previously discussed) is that of the stable cavity attached to the blade. It would seem to be reasonable that the vapor within the cavity is entrained by the violent turbulent closure processes seen at the end of such cavities in water tunnels. In order to maintain the cavity liquid must be evaporated into it from the surface of the flow. The process of evaporation cools the inner bounding surface of the cavity to the degree necessary to cause the heat required for evaporation to flow to the surface. Thus, according to whether the density of the vapour (and hence mass of evaporated liquid) latent heat of evaporation, and entrainment rate are large or not, the cooling effect will be large or perhaps negligible. Other models of the cavitation process also lead to the same conclusion. Thus, for example, it might be argued that in a machine, cavitation occurs in the flow as nuclei that move with the fluid and grow as they encounter the low pressure regions. The rate of growth of such a bubble depends upon the extent to which it is superheated in the low pressure region inasmuch as it is always necessary in a pure liquid to have a certain degree of superheat for a finite growth rate. The actual process as it occurs in a machine probably lies between these extremes. In fact, as Salemann points out, there is indirect evidence to show that the first model occurs when pumping fluids such as cold water and the second for fluids such as hot water that show the pronounced "thermal" effect.

Considerable effort has been devoted by the groups listed in references [24] to [28] (as well as others not listed) to establish the correct similarity laws for cavitating flows with various liquids. Most of the workers take the point of view that in the cavitation process, a certain amount of vapor is formed within the inlet portions of the impeller and that for similar cavitating performance, the ratio of the volume of vapor so formed to the volume of the vapor-liquid mixture must be the same. It is also assumed that the vapor and the bulk liquid are in thermal equilibrium. The evaporation of the liquid is accompanied by a decrease in bulk temperature, and a corresponding reduction of the vapor pressure. The reduction in vapor pressure for a given ratio of vapor to liquid volume will depend upon the particular fluid and accordingly the fluid will be said to have a low or high tendency to cavitate as the depression is high or low respectively. Or conversely, the vapor to liquid volume ratio for a given decrease in vapor pressure may be said to indicate the susceptibility of a fluid to cavitate[25]

The analyses of the depression of the vapor pressure (or reduction in the net positive suction head required for similar cavitation) presented in references [25], [26] and [27] although differing in some details, are all essentially the same; and it can be shown, for example, that the considerations of Jacob[26] are fully equivalent to the following problem of "thermostatics". An insulated beaker with a tightly fitted piston is completely filled with a unit mass of saturated liquid. The piston is then slowly and adiabatically withdrawn until a certain given volume of the vapor is formed. The pressure of the resulting mixture is then to be determined.

This problem thus stated is equivalent to calculating the expansion of the vapor from the point of view of an observed moving with the fluid, provided that there is no friction or heat transfer. The thermodynamic equations for the above problem will now be solved using the symbols of Jacobs.

Let the quality (or mass friction of the vapor) of the mixture be x, and subscripts (1) and (2) denote the beginning and end of the process respectively. Since the process is adiabatic and frictionless, it is reversible and therefore is entropic. Thus

$$S_Q = S_{f_2} + xS_{fg_2} = S_{f1} \tag{10}$$

with the usual thermodynamic notation. It is assumed that the liquid has a negligible coefficient of thermal expanssion, so that the entropy change of the liquid is $s_{f1} - s_{f_2} = c_p n\ (T_1/T_2)$. Writing $s_{fg} = \sqrt{\lambda}/T_2$ one has

$$\frac{\lambda x}{T_2} = C_p n(T_1/T_2) \tag{11}$$

To obtain the connection between the temperature change and the pressure, the Clausius-Clapeyron equation must be used, i.e.,

$$\frac{dp}{dT} = \lambda/V_{fg}T \tag{12}$$

If the temperature change $\triangle T = T_2 - T_1$ is small compared to T_2, the logarithm of Eq. (11) can be expanded, and only the leading term retained when the temperature increment is substituted from the expression

$$\triangle p = -\frac{\lambda^2 x}{V_{fg}TC_p} \tag{13}$$

is obtained, in agreement with Jacob. The depression $\triangle p$ of the vapor pressure necessary to create the vapor is usually expressed in feet of the liquid, i.e., $\triangle P - \delta h/v_f$. Furthermore. the quality x can be elminated in favour of the ratio of volume of vapor to volume of liquid

$$B = \frac{\text{volume vapor formed}}{\text{volume of liquid}} = \frac{xy_{fg}}{V_f} = \frac{V}{V_f} \tag{14}$$

or the ratio of volume of vapor formed to the volume of the mixture

$$V' = \frac{\text{volume vapor formed}}{\text{volume of mixture}} = \frac{xV_{fg}}{V_f + xV_{fg}} \tag{15}$$

resulting in

$$\triangle h = \frac{\lambda^2 V_f{}^2}{V_g{}^2 TC_p} \frac{V'}{1-V'} \qquad (16)$$

and $B = \triangle h/[\lambda^2 v_f{}^2/v_g{}^2 TC_p]$ In there latter expressions the approximation $V_{fg} = V_g$ is also used (17)

If it is now assumed, as Jacobs does, that similar cavitating conditions in a given pump result when V'_2 is the same, then whatever the fluid, the ratio. $\triangle h/(\lambda^2 v_f{}^2/v_g{}^2 TC_p)$ from Eq. (10) should be constant for similar flows to occur. Thus the depression of the vapor pressure, $\triangle h$, necessary to cause the requisite amount of vapor to form, and by supposition, to obtain the same hydraulic performance of the pump, for any fluid can be found if the value of $\triangle h$ is known for any other fluid and if the thermal properties λC_p, v_f, v_g are known for each of the fluids.

On this basis Jacobs is able to correlate fairly well the cavitating performance of a machine pumping liquid hydrogen and liquid nitrogen. Alternatively. Stepanoff chooses the value of B for $\triangle h = 1$ ft as an arbitrary measure of the susceptibility of the fluid to cavitate. Since this quantity shows wide variations from one fluid to another, it is of interest to tabulate several values.

According to the information of Table 1, propane and hydrogen should show little deterioration in performance due to cavitation since the volume of the vapor formed is small. Nitrogen and hot water follow in that order. However, the excessively large value of B for cold water indicates that essentially no depression could actually occur in the pump. Tests conducted with cold water, therefore, give "conservative" results.

Despite the relative success of Jacobs in correlating the performance of nitrogen and hydrogen and that of Stepanoff in his work, it is doubtful that these simple static considerations adequately represent the cavitating process. The constancy of the vapor to mixture volume ratio, V', as a criterion of cavitating similarity was challenged by Salemann on the basis of his measurements using a given pump operated with butane, Freon 11, and water. In fact, he found that if h was determined from the experiments and V' calculated from the thermodynamic relations outlined above, that V'w as a function principally of the vapor pressure of the fluid varying by a factor of about four when the vapor pressure of the flowing liquid is varied by about a factor of 20. It is as a result of this finding that Salemann proposes that cavitation in cold water occurs largely as an attached cavity, whereas the large depression $\triangle h$ necessitated by the higher vapor pressure fluids results in an extensive region of superheated liquid within the impeller in which a more or less homogeneous distribution of separate bubbles occurs. Intermediate states are represented partly by bubbles within the fluid. Thus, the ratio V' need not apply to the entire flow field but only to the liquid adjacent to the cavitation and the restriction of constant V' as a condition of cavitation and the restriction of constant V' as a conditior of cavitation similarity can then be dropped. This ingenious suggestion may be correct. It is also entirely possible that the "static" model of flow sub-cooling outlined above does in fact adequately represent the experimental cases reported. It is significant, however, that neither the rotative speed nor pump size were varied in any of these tests. Moreover, the pumps were fairly small, the maximum flow rate not exceeding 400 gallons per minute. Without a more firm understanding of the mechanics of the cavitating process, one would be hesitant to extrapolate these results to a machine, for example, of ten times the flow rate or ten times the speed. It is difficult to escape the conviction that the time and size scales play an important role, for the average time spent by a particle in the low pressure regions of a centrifugal impeller is only a few milliseconds. With moderate superheats, this time is of the same order as that needed to grow a nucleus in water to an appreciable size[29]. As Plesset and Zwick point out in this series of papers, the rate at which a superheated vapor bubble grows depends upon the vapor density, latent heat, specific heat of the liquid (all quantities found in Eq. (16) and in addition, the thermal conductivity of the fluid. The volume of vapor formed per unit mass will depend upon the individual growth rate of an isolated bubble and in addition upon the number of nuclei present in the fluid. The concentration of nuclei is thought to be strongly dependent upon temperature and may well vary widely from one fluid to another. The history of the fluid may also affect the concentration and, consequently, the cavitating performance[30].

Table 1

CAVITATION SUSCEPTIBILITY PARAMETER B FOR VARIOUS FLUIDS

Substance	*Temperature*	*B for* $\triangle h = 1$ *ft*
Water	70°F	1800
Water	100	353
Water	180	7.87
Water	212	2.08
Water	250	0.66
Water	300	0.16
Butane	70	0.17
Butane	31	0.57
Propane	100	0.008
Propane	44°F	0.92
Hydrogen	46°R	0.01
Nitrogen	138°R	0.40
Oxygen	162°R	0.69
Freon 11	75°F	1.63
Freon 22	91°F	1.28
Freon 113	118°F	1.82

It is quite conceivable that the dynamics of individual bubble growth and distribution of nuclei might in some cases result in the volume of vapor calculated above for the static problem. In any case, it seems

reasonable clear that the problem is much more complex than the simple result of Eq. (13) would indicate, and that to decide between the various possibilities outlined above more experimental work must be undertaken, However, because of the complexity of the phenomena to be investigated, the experimental apparatus should be considerably more simple than that of a centrifugal pump. In addition, it is imperative that the flow be visually observed when cavitating to establish what processes do occur for various fluids, at various temperatures, fluid velocities and cavitation numbers.

Interest in the points just raised above led to the initiation of a modest research program at the Hydrodynamics Laboratory at the California Institute of Technology. A small, recirculating test circuit was constructed that could be filled with water of fluids such a Freeon 113. A transparent working section of glass about one inch in diameter and four inches long is provided to view the flow. Small objects such as plates or discs, are mounted in the working section to form a cavity. The temperature, pressure and velocity can then be varied to see what conditions are necessary to maintain similar geometry of the flow—if indeed that is possible. Of more direct interest is the pressure within the cavity. This is measured by a "U" tube manometer as the difference between the vapor pressure of the bulk fluid, provided by a small closed vessel containing some of the working medium, and the pressure within the cavity itself when the latter is sufficiently clear of froth to permit such a measurement to be made. The vapor pressure bomb and manometer are all enclosed and surrounded by the hot flowing medium in a large stilling chamber just upstream of the working section. Unfortunately, experimental results obtained from the apparatus just described are not yet available. However it is hoped that some information that bears on this problem will soon be forthcoming, and further, that it might contribute to yet another successful symposium still in the future.

Acknowledgement

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Appendix I

Partial Cavitation in a Cascade of Semi-Infinite Flat Plate

I. Theory. Figure 6 shows the cascade geometry. The boundary conditions on the cavity are placed on the upper portion of the chord, which is regarded as a slit in the physical plane. The two halves of the slit are shown slightly separated for clarity. The flow approaches the cascade with magnitude V_1 inclined at the angle to the chord, and ultimately becomes parallel to the chord. It is convenient to define a pressure coefficient based upon the cavity pressure p_c. Thus

$$C_p-(p-p_c)\frac{\rho}{2}v_1^2=K=(v_c^2-v^2)/v_1^2 \qquad (A\text{-}1)$$

On the cavity $p=p_c$ so that $c_p=0$ there. However, from the definition of the cavitation number and the Bernoulli equation

$$p_1-p_c/\frac{\rho}{2}v_1^2=K=(v_c^2/v_1^2)-1$$

V_c is regarded as the characteristic velocity in the vicinity of the cascade blades. Small perturbations with respect to V_c in the horizontal and vertical directions are symbolically written as

$$V=u_1\ v=V_c+u_c v_c$$

were u_c, v_c are much smaller than V_c. At a general point in the flow then.

$$C_p = [V_c^2-(V_c^2+2V_c u_c+u_c^2+V_c^2) \times \frac{1}{V_1^2}$$

$$\simeq -\frac{2v_c u_c}{V_1^2} \qquad (A\text{-}2)$$

The free streamline condition is therefore approximately satisfied by setting u_c equal to zero.

Boundary conditions: The boundary conditions on the velocity function $w=u-iv$ are as follows:

(*a*) $v=v_0=0$ on the wetted portion of the blade (i.e. no flow through the surface).

(*b*) $u_c=0$ on the cavity or $u=v_c$.

(*c*) $w\rightarrow V_1 e^{i\alpha}$ far upstream

(*d*) It is assumed that the cavity is closed. This condition is equivalent to the statement that there is no net source strength in the flow.

It can be shown that conditions (*a*) to (*d*) lead to a unique solution. Since w is an analytic function of $z=x+iy$, the methods of conformal mapping can be used to obtain the solution to the boundary value problem given by Eqs. (A-3).

Mappings: The array of slits in the z plane is transformed onto the real axis and upper half of the unit circle in the ζ plane (Fig. 7) in such a way that the wetted portions of the blade lie along the real ζ axis exterior to the circle. The portion of the slit corresponding to the cavity is the upper half of the unit circle in this plane. The region exterior to the unit circle and in the upper half ζ plane is transformed into the entire z plane. The sequence of mappings used is

$$Z=C\ \ln\left\{\frac{1-t_c e^{i(\pi/2-r)}}{1-t_c e^{-i(\pi/2-\gamma)}}\right\}+e^{-ir}\ \ln \qquad (A\text{-}4)$$

and

$$(t-\frac{b}{2})-\frac{1}{b}=\frac{1}{4}\left(\zeta+\frac{1}{\zeta}\right) \qquad (A\text{-}5)$$

* Model of free streamline flow that do not involve closed cavities are discussed by Parkin in (19).

In the intermediate (t) plane, all of the slits in the z plane are transformed into the real axis of the t plane. A certain length (b) of this axis comprises the cavity portion. Equation (A-5) will be recognized as a Joukowsky transformation which takes this segment into the upper half of the unit circle. The point $t=e^{i(\pi/-2\gamma)}$ corresponds to negative infinity in the z plane and in that plane. The connection between the cavity to spacing ratio c/2 and the parameter b is gotten putting t=b in (A-4) to obtain

$$\frac{C}{2\pi}=\frac{1}{2\pi}\left\{\text{Cos }\gamma \ln (1+b^2-2b\text{ Sin }\gamma)+2\theta_1 \text{ Sin }\gamma\right\} \tag{A-6}$$

where

$$\theta_1=\tan^{-1}[b\cos\gamma/(1-b\text{ Sin }\gamma)]$$

Velocity Function: General procedures are available to solve mixed boundary of the type described by Eq. (A-3) and some of them are mentioned in ([21]). Such an approach is not needed here, for it will be seen that the function

$$W=u-iv=\frac{A}{\zeta+1}+\frac{B}{\zeta-1}+C \tag{A-7}$$

satisfies all of the requirements of the present problem when constants A, B, C are suitably evaluated. The singular terms correspond to sources (sinks) at the leading edge of the hydrofoil and at the termination of the cavity. In addition (A-7) satisfies the here-to-fore unmentioned physical requirement on the flow, namely that the minimum pressure in the flow be on the cavity.

Boundary condition (3a) is clearly satisfied by (A-7) if A, B, C are real since real ζ corresponds to the wetted portion of the blade. Condition (3b) is fulfilled if

$$V_c=\tfrac{1}{2}(A-B)+C=V_1\sqrt{1-k}$$

since on the unit circle $\zeta=e^{i\theta}$ and

$$W\left(\zeta=e^{\theta}\right)=\frac{A}{2}(1-i\tan\theta/2)-\frac{B}{2}\left(1+i\cot\frac{\theta}{2}\right)+C$$

Condition (3c)

$$V_1 C^{-i\alpha}=\frac{A}{\zeta_1+1}+\frac{B}{\zeta_1-1}+C$$

where ζ_1 is the point corresponding to $t=e^{i(\pi/2-\gamma)}$. To evaluate this expression it is convenient to transfer attention to the t plane by means of finding ζ in terms of t from (A-5). After this is done, the complex velocity function in the t plane is found to be

$$W(t)=\frac{A-B}{2}+C+\frac{B}{2}\sqrt{\frac{t}{t-b}}-\frac{A}{2}\sqrt{\frac{t-b}{t}} \tag{A-8}$$

Application of (3c) then leads to a complex expression which, when separated into real and imaginary parts gives.

$$V_1\cos\alpha_1=\frac{A-B}{2}+C+\frac{\text{Cos }\theta_1/2}{2}\left(\frac{B}{l}-Al\right) \tag{A-9a}$$

$$V_1\text{ Sin }\alpha_1=\frac{\sin\theta_1/2}{2}\left[\frac{B}{l}+Al\right] \tag{A-9b}$$

where

$$l=(1+b^2-2\,l\,(\sin\gamma)^{1/4} \tag{A-9c}$$

The remaining condition is the cavity closure condition which is equivalent to the requirement of no net sources in the flow. On the basis of this argument, the continuity law requires the volume outflow to be equal to the inflow or with reference to Fig. (6) one has

$$V_1\cos(\gamma+\alpha_1)=V_2\cos\gamma \tag{A-10}$$

From (A-7) it can be seen that far downstream, $\zeta\to\infty$ and $V_2=C$ there. Thus

$$C=V_1\cos(\gamma+C)/\cos\gamma$$

and all constants can now be determined in terms of k, α and c/2 for a given stagger angle γ. A convenient form for calculation obtained after some manipulation expresses the cavitation number in terms of the parameter b:

$$\sqrt{1-k}=\cos\alpha-\sin\alpha\,\frac{l\cos\alpha}{l^2-(1-b)\sin\alpha}\cdot\cdot$$

$$\times\frac{(1-l^2)\cos\gamma-\sqrt{2l}\left(1-\dfrac{1-b\sin\gamma}{l^2}\right)^{1/2}\sin\gamma}{(1+l^2)\cos\gamma-2\sqrt{2l}\left(1-\dfrac{1-l\sin\gamma}{l^2}\right)^{1/2}\cos\gamma}$$

where l is given by Eq. (A-9)

Discussion. Equations (A-6) and (A-11) constitute the solution of the problem. For given values of b the cavity-spacing ratio is calculated from (A-6) and corresponding values of k are then determined from (A-11). Typical results are shown in Fig. 8 for various values of γ at an angle of attack equal to six degrees.

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Remarks on Cavitation in Turbomachines

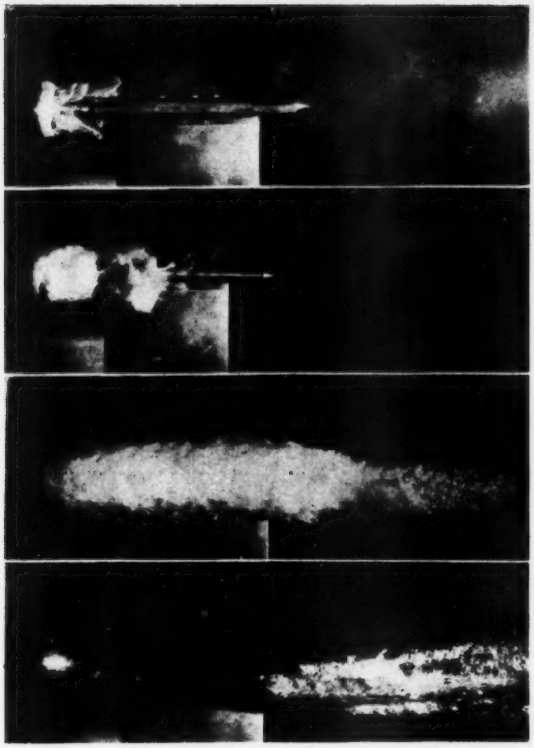
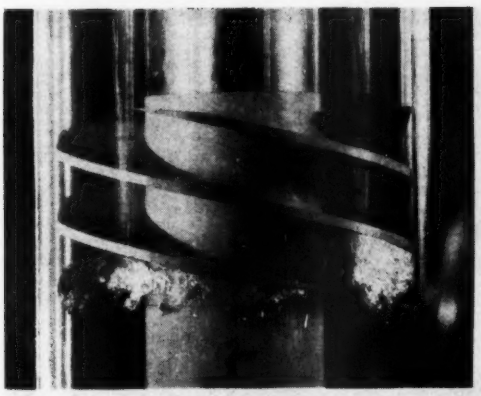


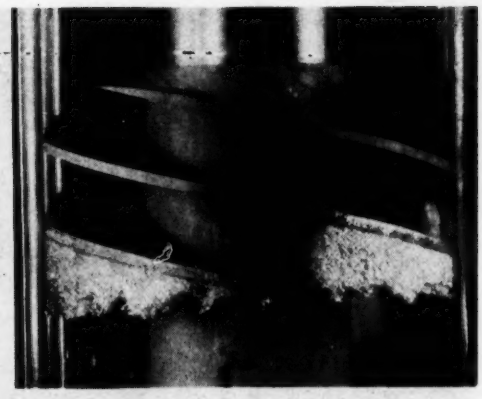
Fig. 1. Progressive Development of Cavitation on a Sphere

Fig. 2. Tip Vortex Cavitation on an Axial Flow Pump

Fig. 3. Development of Cavitation in a 12° Helical Inducer for a Flow Rate Coefficient $\Phi = 0.12$. (These Photos are not Strictly in a Sequence Since They Are Taken at Different Times and Rotative Speeds).



k = 0.09



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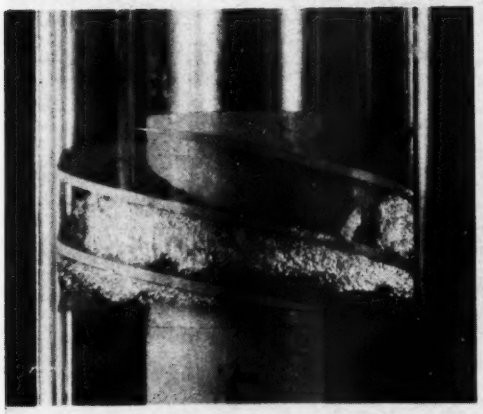

k = 0.023

k = 0.02

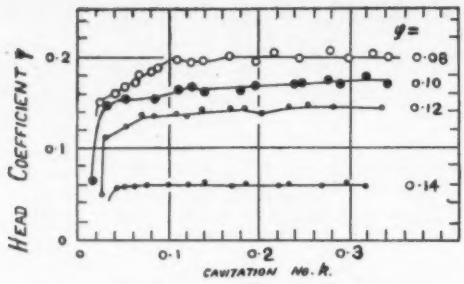

Fig. 4 Head Coefficient vs Cavitation Number at Various Flow Coefficients for an Inducer Type Impeller. The tip blade angle is 12°, the solidity is 2.5 and the hub to case ratio is 0.5.

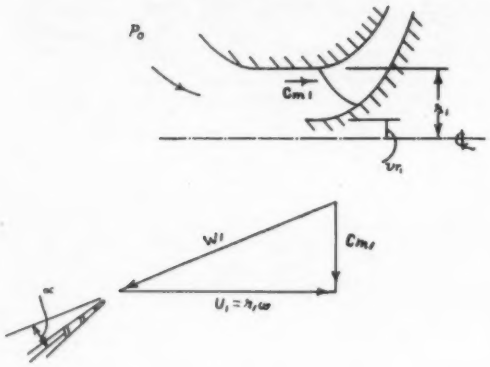

Fig. 5 Velocity Triangles at the Entrance to an Impeller.

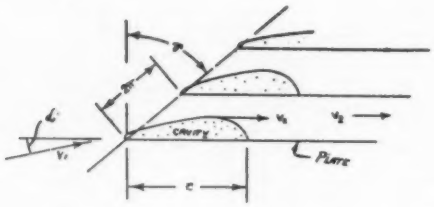

Fig. 6 Sketch showing Partial Cavitation in a Cascade of Semi-Infinite Flat Plates.

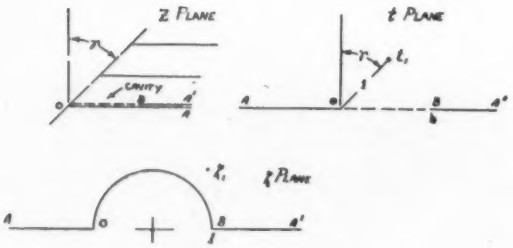

Fig. 7 Sketch Showing Various Transformation Planes.

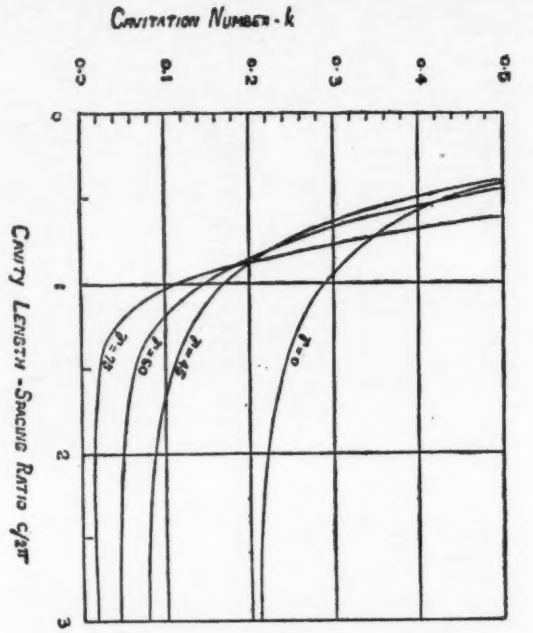

Fig. 8 Cavitation Number vs Cavity Length to Spacing Ratio for an Angle of Attack of 6 degrees with Various Values of Stagger Angle γ.

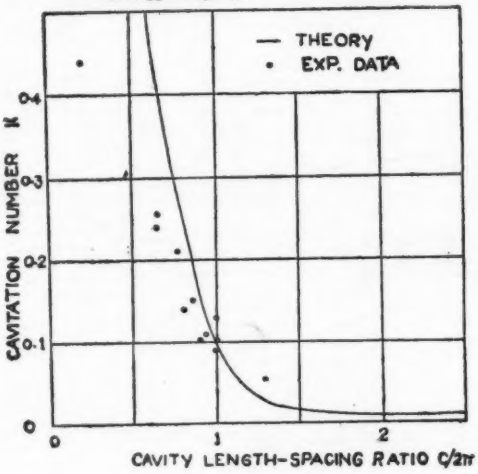

Fig. 9 Cavitation Number vs Cavity Length Compared with Experimental Observations on a Helical Pump Inducer Impeller.

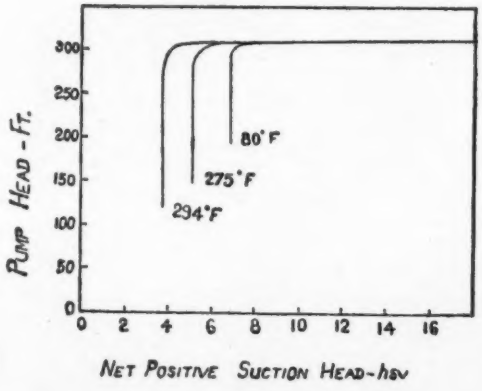

Fig. 10. Pump Head vs Net Positive Suction Head at Constant Flow Rate and Speed for Water at Various Temperatures [after Stahl (10)].

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Notations

A,B,C	—	Constants
A_1	—	Inlet area (ft^2) of impeller
b	—	Parameter
c	—	Chord (ft)
c_m	—	Meridional velocity (ft/sec)
c_p	—	Specific heat at constant pressure (Btu/lb°R)
g	—	Gravitational acceleration
h	—	Impeller head (ft)
h_{sv}	—	Net positive suction head (ft)
$\triangle h$	—	Reduction in vapor pressure (ft)
i	—	$\sqrt{-1}$
k	—	Cavitation number $= (p_1 - P_1)/\frac{1}{2}\rho_1 W_1^2$
N	—	Revolutions per minute
p	—	Pressure
Q	—	Flow rate, cubic feet per second (except in Eq. 3)
r	—	Radius (ft)
s	—	Entropy Btu/lb °R
S	—	Suction specific speed (Eq. 3)
T	—	Absolute temperature (°R)
t	—	Variable
u	—	Velocity (ft/sec)
v	—	Velocity (ft/sec)
V	—	Volume (ft^3), velocity
V'	—	Ratio of vapor volume to volume of mixture
w	—	u-iv velocity function, relative velocity to impeller
x	—	Coordinate, quality of vapor-liquid mixture
y	—	Coordinate
x	—	x+iy
α	—	Angle of attack, measured between inle relative velocity and chord line
γ	—	Stagger angle
γ_2	—	Complex variable
λ	—	Latent heat of evaporation (Btu/lb)
ν	—	Ratio of hub diameter to inlet diameter
ρ	—	Density (slugs/ft^3)
σ	—	Thomas' cavitation parameter (Eq. 1)
μ	—	Flow coefficient, meridional velocity divided by tip speed of impeller. Thus $1 = Q/A_1 r_1$
Ψ	—	Head coefficient, $h/(u^2/g)$ u=r.
ω	—	Angular velocity (rad./sec)
1	—	Pertains to inlet diameter
2	—	Pertains to discharge diameter
c	—	Characteristic velocity, perturbation quantities
f	—	Pertains to liquid properties
g	—	Pertains to vapor properties
in	—	Refers to meridional quantity
p	—	Pressure
v	—	Denotes vapor pressure.

Speed Control of a Prime Mover

By G. FORREST DRAKE

The automatic control of machines and processes for the purpose of maintaining some quantity constant at a desired value has achieved a position of great importance in modern technology. The continuing establishment of higher standards of performance and the increasing complexity of the systems to be controlled have compelled the attention of large numbers of engineers and scientists and the application of advanced analytical methods to the problems thus presented. So rapid and great has been the growth of the need for solving such problems that a new field of engineering comprising thousands of engineers and scientists has evolved in the last decade. This is Control Engineering, and its basic tool is servomechanism theory.

Because differential equations comprise the minimum mathematics pre-requisite to a working understanding of servomechanism theory, many people who are concerned in a very direct and practical way with the manufacture, operation and maintenance of automatic control systems find themselves at a loss to understand what is going on even in a very general and qualitative way. This paper will attempt to bridge the gap between the control theorists and those who, with limited technical training, are nevertheless concerned with the same mechanisms and problems. We will limit our efforts to a very specific area of automatic control—that of the control of the speed of rotating machines.

Our intention will be to cover in an elementary manner the facts necessary for a general understanding of prime mover speed governing. No prior knowledge of the field or even of the elementary mechanics involved will be assumed. Our approach must of necessity be reasonable explanation rather than rigorous proof of the technical matters discussed, and we must ask that many statements be accepted on faith.

Little time will be spent on details of either governors or prime movers, for these are covered in other bulletins. Rather, the important factors in the complete system comprising prime mover, load, and governor will be discussed in a general way so that the characteristics important to control results may be recognized.

The Prime Mover has been defined as a mechanism which converts heat or hydraulic energy into mechanical power. This definition includes water wheels, steam engines and turbines (with their boilers), gas turbines, gas, gasoline, and diesel engines; it probably should be extended to include wind turbines or "windmills." It does not include electric or compressed air motors although these are at times controlled by standard governing devices.

All of these prime movers, varied as they are in construction and principle of operation, have at least two things in common; first, they derive their power from the flow of some Energy Medium (such as gas, fuel oil, or water); and second, they are rotating machines. In the usual case, their speed is controlled by controlling the rate of flow of the energy medium, and thus the power developed. That the speed can be controlled by this method follows from consideration of the mechanical principles involved.

MECHANICS

Most of us have a feeling for Force. Physically we sense it as the muscular effort required, for instance, to hold a pail of sand off the floor. The force in this example is due to the attraction called Gravity which exists between all particles of matter. We measure such a force in pounds weight or more briefly, pounds. The weight of a body is therefore the force, acting toward the earth, which results from the gravitational attraction of the body and the earth for each other. The property of a body which causes it to have weight, is its MASS. The mass of a body is a measure of the quantity of matter of which it is composed. Work is done when a force acts through a distance. Thus if we simply hold the pail against the force of gravity we do no work in the mechanical sense, even though we may suffer muscular fatigue. If we lift the pail, however, we do work on the pail and increase its Potential Energy. Thus work and energy are seen to be equivalent, the energy of a body being a measure of its capacity to do work. In this case, by reason of its position, it can do work on the supporting hand if a allowed to descend again to the floor. Compressed gas or a compressed spring similarly have potential energy.

Mass imparts another important characteristic to a body. Anyone who has ever thrown a rock is aware that an effort is required to get it in motion. The property of the rock that requires this effort is the Inertia of its mass. This is exhibited in the tendency of a body to remain at rest, or if already in motion, to continue in a straight line at constant velocity unless acted upon by an external force. This is Newton's first law of motion. When such as external force acts on a mass, it imparts to it an Acceleration which is Rate of Change of Velocity. Velocity in turn is Rate of Change of Position.

We are primarily concerned with rotating machines in which the masses are forced to follow circular paths, rather than straight lines. In such a case, even if the machine is rotating at constant angular

velocity, that is, constant rpm, a force must be exerted to compel the particles to follow the curved parth. This is the so-called Centripetal force, equal and opposite to the Centrifugal force which is utilized in the speed sensing ballhead to be described later; in a flywheel or other rigid rotating machine element the centrifugal force simply stresses the material. When a machine is properly balanced the centrifugal forces cancel each other and, if not excessive, are of no further concern. However, the forces required to accelerate the particles tangentially, that is, to increase the speed of rotation, add up to a Torque which is proportional to the angular acceleration times the Moment of Inertia of the machine. Torque is a measure of the turning effort exerted and is proportional to the product of a tangential force and the length of the lever or crank arm on which it acts. The moment of inertia is a measure of the effective inertia of the mass of the machine in rotation. It is commonly expressed as WK^2 (usual units, pound feet squared), and the angular acceleration in rpm per second. This relationship between torque, moment of inertia, and acceleration is for it permits us to compute the acceleration of rate of change of speed of a prime mover for which we know the WK^2 if we also know the Net Torque. This torque is determined as follows.

Torque times speed of rotation is Power. Therefore if speed is held constant, torque and power are proportional. When a prime mover is running at equilibrium at a given speed, the Required Torque (corresponding to the load power) is exactly equal to the Output Torque. This Output Torque is equal to the Developed Torque resulting from proper flow of the energy medium into the prime mover less the torque required by prime mover auxiliaries, friction and other internal losses in the machine. Under these conditions, the Net Torque, which is the difference between output and required torques, is zero and no acceleration or change of speed can result.

However, when the load (and therefore the required torque) is suddenly decreased, the output torque will momentarily remain unchanged and there results an excess or net torque available which immediately produces an acceleration of the machine and the speed increases. The magnitude of the total speed increase (for a given initial net torque) depends upon the rapidity with which the power developed can be changed (by the governor) to the new proper value, and upon the WK^2 of the machine. During the time that the power developed is greater than the load, an excess of energy is produced which must go into increased Kinetic Energy of rotation.

Kinetic energy is energy stored in a body by reason of its motion; in this case its rotation. It is proportional to the WK^2 of the rotating mass times the square of the speed of rotation. Therefore, the larger the WK^2 the smaller the speed change necessary to absorb a given amount of excess energy. Similarly, if load is suddenly added, the momentary decrease in speed is less for the greater WK^2, all else being equal. However, other important factors must also be considered in selecting the flywheel for a given unit, of which more will be said later. Left to itself, the prime mover will not return to its original speed following a load change. To accomplish this it is necessary on overspeed, for example, to decrease the output torque below the new required torque, thus producing a Deceleration or Negative Acceleration. It is the function of the governor to do this automatically and to re-establish a torque balance when the machine is again at the desired speed.

SPEED CONTROL METHODS

It has been noted that the usual way to control the speed of a prime mover is by controlling the power developed. There are, by implication, other ways that should be mentioned. For instance, the flow of the energy medium may remain constant and its effectiveness, that is, the efficiency of the machine, varied to compensate for changing loads. An example of this method of control is the impulse type hydraulic turbine in which that portion of a constantly flowing water jet which is allowed to strike the buckets of the wheel, may be varied by positioning a "deflector hood". Another is the wind turbine in which the pitch of the blades is adjusted to maintain speed under varying conditions of both load and uncontrolled energy medium flow.

Where it is possible to control the load completely, speed may be maintained by adjusting the load torque to equal the torque developed rather than the reverse. Such a system is in common use in the constant speed aircraft propeller. In this case the engine power is set at a chosen value and the governor maintains the desired speed by varying the propeller pitch to provide the required load under changing conditions of aircraft speed and air density.

What has been said indicates the manner in which the speed of prime movers can be controlled. It does not establish the necessity for automatic control in the face of the fact that thousands of prime movers are operating without it. Obviously, automatic control must be much more important in some cases than in others. The differences responsible for this fact are to be found in the characteristics of the prime movers and their loads.

Although most of that which will follow is pertinent to the control of prime movers in general, it is convenient to be more specific when presenting examples; in such examples, therefore, we will assume internal combustion engines as prime movers, since they are relatively well-known to nearly everyone.

PRIME MOVER CHARACTERISTICS

The ultimate purchaser of a prime mover is interested in its performance primarily with regard to its efficiency and service life; in general, he assumes that satisfactory control will be provided. The control engineer, however, is concerned with other steady-state characteristics, and even more with its dynamic characteristics. The sort of information he needs has not been readily available in the 'past, and considerable

effort has been expended in securing such data and in enlisting the cooperation of the prime mover manufacturer in providing it. This becomes an easier task as the importance of these data in the design of a satisfactory control system becomes more generally recognized.

One very important steady-state characteristic is the manner in which the output torque varies as speed changes, the gate or throttle remaining fixed. If this torque decreases with increasing speed, as is the case in varying degree with all turbines and naturally aspirated gas and gasoline engines, the unit tends to be "self-regulating"; that is, if the required torque remains constant, the net torque resulting from the change in speed is in the direction to oppose further change (Fig. 1). This characteristic is sometimes referred to as Damping. It can also be present in the load, as will be seen later. A large amount of damping in the controlled system makes the job of stable control easier. However, the speed deviations resulting from large load changes at fixed throttle even in such units are much too large to be tolerated, and a governor is required to maintain acceptable speed. Nevertheless, if only small, or in any event slow, load changes are experienced, as in a marine drive, speed can be held very satisfactorily by manual positioning of the throttle.

In the case of a gasoline engine with gear driven supercharger, it is possible to have a condition in which with increased speed the blower more than compensates for increased induction losses, and increased output torque may result (Fig. 2). Thus a change in speed produces a net torque in the direction to further change the speed, and Negative Damping results. Manual control of such a system is very difficult, and with automatic control stabilizing forces must be introduced by the controlling governor in greater degree than would be required for the first example.

If injector peculiarities are neglected, the diesel engine will, for a given fuel rack setting produce a torque which is substantially independent of speed (Fig. 3)

It may be of interest to point out that the self-limiting characteristics of the unsupercharged or normally aspirated gasoline engine will, in most cases, establish a balance between the torque required to overcome friction and windage losses and the torque developed in the unloaded engine at full throttle at a speed sufficiently low to prevent self-destruction. This is also true of modern hydraulic turbine designs. However, it is not so in the case of the diesel engine. Therefore in applications where there is a possibility of sudden loss of load, a governor, at least as a maximum speed limiting device, becomes a necessity for the diesel.

In the preceding discussion we have considered only the steady-state characteristics of a few typical prime movers at fixed throttle. As soon as the governor takes charge, the fixed throttle condition disappears and many other factors become important. We also know that in practice a truly "steady-state" condition never exists and therefore, in the presence of continually changing conditions we must take account of the transient or dynamic characteristics of the prime mover and its load.

One of the most important considerations from a control stand point is the time which elapses between the movement of the gate or throttle to a new position and the development of torque at the corresponding new level. This lag in response varies greatly between prime mover types and is one of the principal reasons for the differences between them in control results.

In the case of a hydraulic turbine the most serious lag results from the water itself. This water has mass and inertia and there is a limited force (the head) available to accelerate it to a new flow rate. Therefore, if load is suddenly added, it does no good and may do harm to open the gates faster than the water can accelerate. During this period of deficiency in torque developed, the wheel will underspeed. Similarly load rejections, time is required to decrease the flow rate; in this case because of the pressure limitations of the conduit. The water wheel case is further complicated because, as the gate opening is decreased there is, initially, little change in flow rate (because of the time lag mentioned), and therefore the velocity of the water impinging on the runner momentarily increases with a resulting increase instead of a decrease in torque.

In internal combustion engines the time lag has three components: the lag in the manifold (if a gas or gasoline engine), the dead time between the charging of the cylinder with fuel and its conversion to torque, and the time required for all the cylinders which simultaneously contribute to the developed torque to be firing at the new level. The last is present in all engines in which more than one cylinder at a time is producing torque and is apparent when one considers that even the charging rate instantaneously follows throttle movement and cylinder instantaneously begin charging at the new level, some are currently firing at the old level and until all of these have been replaced by cylinders charged at the new level, full torque development at the new level will not have been achieved. In the case of a diesel engine this is the only time lag of any magnitude and amounts to perhaps one-third revolution in the average case.

The "dead time" varies greatly and is least in the usual diesel and greatest in the 4 cycle gas engine. This accounts in no small measure for the differences in control of dual fuel engines on oil and gas. In a 4-cycle normally aspirated gas engine, fuel is admitted during the suction stroke and does not appear as torque until nearly a full revolution later. In a similar 2-cycle engine, fuel enters near bottom dead center and begins to appear as torque shortly after top dead centre or after a dead time of approximately two-third revolution. Since the diesel adds fuel at the beginning of the power stroke, substantially no dead time exists. Dead time may be reduced to a minimum in a gas

engine by injecting gas at high pressure at the same point in the cycle. Such gas engines rate in controllability very close to diesels.

It should be noted that the two time lags just mentioned were described in magnitude in terms of revolutions of the crankshaft and that they therefore decrease in true time as the speed of rotation increases, being in fact inversely proportional to speed, all else being constant.

The first time lag mentioned was that due to the volume of the intake manifold and the compressible nature of the fuel and is subject to perhaps the great est variation of all. The manifold, by its accumulator action, makes it impossible for the pressure ahead of the gas intake valve to follow quickly changes in throttle position. This pressure then gradually attains its new equilibrium value after an instantaneous change in throttle position, and the developed torque is correspondingly delayed in reaching its new level. This lag is very frequently the largest of the three mentioned, although in later designs steps have been taken to decrease it. The most improvement, short of high pressure injection, is secured by placing a control valve at each cylinder with minimum volume between it and the gas intake valve.

These delays affect control adversely in two ways. First, it is obvious that the speed deviations resulting from load changes will be greater the greater these time lags even though the throttle were instantly moved to the proper new position, because of the resultant delay in making the needed corrective change in the developed torque; second, it is necessary to increase the compensation setting of the governor in order to secure stability of a system with such lags, which results in decreased speed of throttle movement on speed change and still greater speed deviation before recovery. For these reasons, the quality of control, that is, the accuracy with which the desired speed is maintained, even at design load conditions, will be poorer as the time lags mentioned become greater, all else remaining the same.

Turbo-supercharging is important because of its increasing use on both diesel and gas engines. In spite of the large time lag introduced by the turbo-charger, which results from its slow acceleration, steady-state stability is practically unaffected by its presence. This is for the reason that, for the small fuel changes necessary to maintain speed under steady-state load conditions, there is adequate air available without change in supercharger speed, and the transients are over before any significant change in that speed can occur. For large load increases, however, the story is quite different. In such cases the air needed to burn the fuel required by the new load may not be available until the turbine and compressor have had time to speed up, with the result that serious smoking or even a stalled engine may result. This is a problem outside the scope of a simple speed governor and additional controls are required to limit the fuel to an allowable amount. Under this condition full load torque is unavailable instantaneously either for carrying load or accelerating from a lower speed.

Another engine characteristic of great importance to control is its moment of inertia, WK^2, or "flywheel effect". This must include, of course, the WK^2 of its driven machine, such as an alternator. As mentioned earlier, any difference between torque developed and torque required by the load must go into acceleration or deceleration of the machine. For a given torque difference this acceleration, positive or negative, is inversely proportional to the WK^2; that is, with twice the WK^2 a given engine will change speed only half as fast. In general, therefore, greater WK^2 allows more time for the throttle to move in the corrective direction and permits maintaining speed with less deviation. But WK^2 costs money, and a compromise value is usually selected. Furthermore, it is possible to get worse control results by adding WK^2 is such addition brings the natural frequency of a synchronous machine closer to resonance with some forcing frequency , as will be mentioned later.

It should be noted here that upon full load rejection, and with fuel completely off the only torque available to decelerate the engine is that of friction, windage and whatever other internal loads the idling prime mover may have. Therefore it may take an appreciable time for it to return to speed, and the greater the WK^2 the longer this time.

The addition of WK^2 is not an unmixed blessing from the control transient standpoint even for normal load variations since the speed of recovery from a disturbance is adversely affected under such conditions as well. This implies compromise, in the usual case, between speed deviation and time of return to speed, and the choice of WK^2 may frequently be determined by which of these is the more important in the given application.

Few engine characteristics have more effect on control stability than the relationship between developed torque and throttle position, and careful attention to the design of this detail is well repaid in performance. In the usual diesel injector this is an approximately linear relationship, and this together with the small time lag in response, provides a system relatively easy to control. At the other extreme is a gas or gasoline engine with fuel controlled by an ordinary butterfly valve. In such a valve, the initial opening movement effects a relatively large change in fuel flow. In this region of light load it is quite usual to require more compensation or stabilizing effect in the governor than is required at full load. Since this stabilizing effect is proportional in magnitude to the movement of the governor servo, the small movement required at low load and the large movement at high load for a given change in developed torque produce a result just the reverse of that desired. Considerable improvement can often be secured by properly designing the linkage so that angularity may be taken advantage of to given large governor movement for small throttle movement at no load (Fig. 4). Another solution is to use a valve with ports contoured to give the pro-

per area change for a given movement throughout the stroke. By these means, we can if we wish even provide greater governor movement and therefore more compensation at light loads than at full load.

This feature may be especially desirable in certain engine designs in which the fuel/air ratio is optimum only at high loads and which, for this reason or because of lower compression pressures, tend to run erratically at light loads. This chararteristic is present in almost all gas engines to some degree and contributes its share with the factors mentioned above to the generally poorer control obtainable with gas and dual fuel engines on gas as compared with diesel operation at light loads. Improvements in dual fuel operation on gas at light loads have been observed to result from increasing the pilot fuel used. This is probably due to the fact that, since a larger percentage of the load is carried on the more uniform oil, the effect of fluctuation in the energy supplied by the gas and the time lags involved in the gas control are reduced, and that increased pilot fuel tends to provide more uniform combusion of the gas that is used. At full load which approximates the condition for which they are designed, all engines tend to run better and control easier. The exception is usually due to resonance appearing under this load condition either in synchronous machine or in criticals within the engine itself which, passed on to the governor through its drive, cause undesirable speed variations and seemingly erratic control results.

LOAD CHARACTERISTICS

Although much has been said about prime movers, it is not intended to imply that the load characteristics are of lesser importance in determining the over-all control results. Unfortunately, however, usually nothing can be done about them; they must be accepted as they exist and taken into account in the control design. For instance, consider the type of load exemplified in the marine propeller, the airplane propeller, and the fan or blower. It is a fair approximation to say that the torque required by such a load varies directly as the square of the speed; that is, a 1% increase in speed would require a 2% increase in engine torque. Such a load is considered to have high positive damping; in fact, about the highest that is normally encountered. Combination with a prime mover such as a steam turbine which also has positive damping results in a very stable system with a high degree of "self-regulation", as described before (Fig. 5)

A direct current generator with constant excitation generates a voltage directly proportional to speed. Such a generator supplying a resistance load would have a power output proportional to the square of the speed and a required torque directly proportional to the speed. This load would also have positive damping but to a lesser degree than propeller and blower. However, if this generator were equipped with voltage regulator sufficiently fast in action to compensate for the changes in speed and maintain the voltage constant, the results would be quite different. In this case the power output would remain constant since the voltage is not permitted to change, and since power is proportional to speed times torque, an increase in speed would be accompanied by a decrease in torque, and negative damping would result. If driven by a diesel which has substantially no damping, the result would be an unstable system and the controlling governor would have to exert a much greater stabilizing influence than in the case first described (Fig. 6).

The above example serves to illustrate another fact which is frequently overlooked; that is, that other controls such as voltage and pressure regulators operating on the same controlled system can and often do have a definitely unstabilizing influence. In such cases it may be that any one of the several control systems operating by itself is perfectly satisfactory and completely stable, but combined operation is impossible. The usual solution to this problem is to select the quantity for which the most accurate control is desired, such as the speed, and make its control as fast and accurate as possible, assuming all other quantities constant; then the controls of the secondary variables are deliberately made sufficiently slow so that this assumption is substantially true during the time of the speed transient.

The loads just described are characterized by changes of torque with speed, but they contain no equivalent to the Synchronizing torque which exists in an alternator supplying a synchronous load. In this case torque equilibrium is not re-established upon return to synchronous speed only; there is an additional torque which is dependent upon the position of the alternator rotor relative to the synchronous field, as though the load were driven through a torsion spring. This type of load will be discussed later and it is desired at this time only to point out that, because of this synchronizing or "spring" torque and the rotating mass, a natural frequency of oscillation about the equilibrium synchronous speed position exists. This fact makes an entirely different problem of the control of an engine driving an alternator on synchronous load as compared with the same machine off the line or supplying a non-synchronous load such as heaters or lamps. Even with the later load, if other alternators are paralleled with the unit in question a synchronizing torque exists between them.

SPEED GOVERNORS

In the foregoing discussion we have concerned ourselves primarily with the effect of engine and load charactristics on governing rather than with the governors themselves, to which we now turn our attention. The use of the word "precision" as it is used in connection with speed governors implies that there are governors which are not precise, or that there are varying degrees of precision in governor control. This is, of course, true. It is also true that for many purposes, such as limiting controls in the automotive field, relatively crude and insensitive governors are perfectly adequate for the job required of them. An outstanding example of this type of governor is that used to control the throttle on small engines such as those used to power lawn mowers. In this case a flat vane is placed in the blast from the cooling fan and so pivoted and connected to the throttle valve that an

increase in the air blast resulting from increased speed moves the vane against a light spring in the direction to close the throttle (Fig. 7). This represents about the ultimate in inexpensive governing, yet it is completely adequate for the application. Our interest, however, lies with the job which requires high accuracy of control and the availability of considerable force to operate the throttle. Such a problem does not yield to such a simple and inexpensive solution.

It is apparent from what has gone before that a speed governor must include at least two components; a speed sensing element and a device to operate the throttle. In the simplest governors, as we have seen, these may be one and the same. Where considerable force is required, however, an additional device called a servomotor is required. This servomotor, capable of exerting the required force, is controlled by a speed sensing element which in itself may have very little energy available at its output. A number of means which are used for sensing speed changes and actuating the gate or throttle will be mentioned.

A speed senser of long standing consists of a centrifugal pump in which the dead-end or zero flow pressure is taken as an indication of speed (Fig. 8).

Another uses a constant displacement pump discharging through a fixed orifice, the pressure drop across the orifice being indicative of speed (Fig. 9)

A number of electrical schemes have been used including a small d.c. generator with permanent magnet field for which the output voltage is proportional to speed (Fig. 10)

A variation of this scheme which eliminates commutator and brushes uses a small permanent magnet alternator, the output of which is rectified, the d. c. voltage being used as before (Fig. 11).

A third electrical scheme uses the permanent magnet alternator, but feeds its output into a frequency sensitive network which provides a signal approximately proportional to the deviation in speed from an equilibrium value (Fig. 12).

All of the above have their advantages and disadvantages which determine their fields of application. Woodward has considered and following special purposes used most of them, but the speed sensing device preferred for its sensitivity, ruggedness, and the usefulness of its output is the centrifugal ballhead.

This mechanism is probably the oldest and certainly one of the simplest of speed sensing devices. It depends for its operation upon the fact that a force is required to compel a mass to follow a circular path, as mentioned earlier. This force is proportional to the square of the speed of rotation and to the first power of the distance of the mass from the axis of rotation. In its best known form the ballhead consists of a pair of weights, usually spherical, at the ends of two arms pivoted near the axis of rotation in such a way that flyweights can move radially in a plane through the axis. Additional links are attached to the arms and collar about the axis to form a parallelogram configuration. Thus, when the weights move outwardly, the collar moves up (Fig. 13). Since the centrifugal force always acts at right angles to the axis of rotation, it exerts a torque about its pivot equal to the product of the force times the vertical distance of the ball below the pivot. This torque is opposed and, if no other forces are present, must be balanced at equilibrium by the torque of gravity, which is equal to the weight of the ball times the horizontal distance to the pivot (Fig. 14). Thus as the speed increases the centrifugal force torque arm and increasing the gravity torque arm until equilibrium is reached. This results in a unique equilibrium position of flyball and collar for each speed of rotation. In the direct mechanical governor this collar is connected to the throttle so as to close it as the flyweights move outward. Early steam engine governors were of this type. It should be noted that the unique relation between speed and position of ballhead and collar no longer exists if friction is added to the system. This is for the reason that as speed increases from an equilibrium condition the centrifugal force torque must reach a value equal to the gravity torque plus that due to friction before movement results. Similarly on decreasing speed the centrifugal torque must go down to a value equal to the gravity torque minus that due to friction. The result is a so-called "dead band" or region in which the speed may wander without producing a corrective motion of the throttle. Efforts to minimize this dead band resulted in the huge cast iron flyweights common to early mechanical governors.

From the above it is apparent that for greatest sensitivity the force available from the ballhead upon a small speed change should be large relative to the force required for control. There are two approaches to this problem; the friction of the entire system and the force required for control may be minimized, or the force available from the ballhead upon a small speed change may be increased. The second is accomplished when the size of the ballhead is increased or when a small ballhead is run at high speed. The first implies careful construction to reduce friction, and removal of the throttle load to a servomotor controllable by a ballhead output of low power level. Over the years the centrifugal ballhead has progressed through a great number of design configurations to reach the form currently used in Woodward governors (Fig. 15). As used in the governors for internal combustion engines it usually consists of two weights with their centers of mass approximately the same distance from the axis of rotation as the pivots about which they swing. So-called "toes" are arranged substantially at right angles to the body of the fly weight (they aren't "balls" any more) in such a way that as the weight moves toward and away from the axis of rotation, the toes convert this motion to an axial movement of Pilot Valve or Speeder rod through a suitable thrust bearing. The centrifugal force is opposed and balanced by the force exerted by a compressed Speeder spring instead of gravity. In some designs the toes are on the center line, and since they move in an arc, some

sliding with resultant friction takes place. In other designs the toes are offset and contact the thrust bearing on a line at right angles to their plane of movement so that the arcuate movement is converted into a slight rotation of the thrust bearing with a minimum amount of sliding friction (Fig. 16). Friction is further reduced in the more sensitive designs by the use of antifriction bearings at the pivots. These precauions are taken to reduce the Dead band which results from friction as described earlier.

It might be noted here that complete elimination of friction from the ballhead produces an undesirable result. This is for the reason that the flyweight-speeder spring combination, forming a pendulum system, will tend to oscillate after a disturbance, giving a false indication of speed deviation. It is desirable to have sufficient damping in the system so that it will return to equilibrium with no more than one very small overswing. It would appear that this might best be done by utilizing a dashpot to provide viscous damping for a frictionless ballhead, inasmuch as this would give the required damping without producing dead band. A new Woodward ballhead for the cabinet actuators is, in fact, designed this way, but primarily for different reasons. In this case it was felt that this inherently more expensive construction was justified in order to achieve maximum performance over a long period without lubrication. In the smaller governors lubrication is inherent in the design, and this construction is unnecessary, since the amount of friction required for satisfactory damping is so small that the best we have been able to accomplish in the reduction of friction in the pivoted ballhead still leaves enough for adequate damping and is at the same time sufficiently low to provide an acceptably small dead band.

As stated above, the force necessary to restrain tne flyweight, in this case by loading of the flyweight toe, increases with the square of the speed and directly with the radial distance of the flyweight center of gravity from the axis of rotation. Thus, at a given speed, the force will increase as the flyweight is permitted to move out. What is called the "flyweight scale" is the rate at which this force increases with movement, referred to the flyweight toe. Thus if at some speed a movement of ·010" at the toe produces a force change of 0.2 lb. the flyweight is said to have a scale of 20 lb/in., which for the usual ballhead with two flyweights is 40 lb/in ballhead scale. For a given ballhead this is different at each speed, in fact varying proportionally with the square of the speed.

It should be noted that as the flyweights move out, the speeder spring is compressed. Since the speeder spring also has a scale, this results in an increase in the force opposing flyweight movement. If this increase is less than that of the ballhead for the same movement, instability results. In other words, if a condition of equilibrium exists which merely requires ballhead and spring forces to be equal, and a slight displacement of such a combination takes place, the new forces of ballhead and spring are not equal and the net force is in the direction to continue the motion. Such a ballhead and speeder spring combination will, upon varying speed over the necessary range, snap quickly from one extreme position to the other. This characteristic may be designed into an overspeed trip, but it is useless in a regulating governor.

If, on the other hand, the scale of the speeder spring is appreciably greater than that of the ballhead, a stable system results; one having a unique equilibrium position for each speed within the range of movement of the flyweights. Regulating governor ballheads are so designed. By properly choosing the ratio of speeder spring and ballhead scales a wide range of sensitivities or movement of pilot valve for a given speed change can be had. The best choice, however, depends upon the nature of the system which the governor is to control.

Thus far no mention has been made of an additional force factor which must be taken into account in ballhead design, namely the hydraulic reaction force exerted on the piston type pilot valve due to flow of oil through the ports. This force, for the usual range of port openings, is in the direction to re-center the valve and varies approximately linearly with the valve opening. Thus this force also has a scale which, in the case of the directly connected valve, is additive to the spring scale, and if not taken into account may produce a ballhead of much lower sensitivity than anticipated. The magnitude of this force is a function of the flow through the valve and the pressure drop across it and is the principal limiting factor in the determination of the size of valve that can be satisfactorily actuated by a ballhead of given size.

This implies that control of large oil flows, as required in water wheel governors, could not be handled directly by a ballhead of reasonable size. This problem is solved in cabinet actuators and water wheel governors by the use of an additional valve. This valve, capable of controlling large oil flow, is arranged so that it takes a position proportional to the speeder rod position, but actuated by a hydraulic force many times greater than that available from the ballhead (Fig. 17)

As has been mentioned, the ballhead force and scale vary with the square of the speed. This suggests that to design a speeder spring for wide range operation may be something of a problem, since it is desirable to have a nearly linear relation between speed and speed setting and since for proper operation the ratio between spring scale and ballhead scale should not vary too much. The problem has been quite satisfactorily solved for speed ranges of six to one and, in special cases, even higher, by the trumpet shaped or conical spring. This spring has a non-linear load deflection curve (Fig. 18) since it is so wound that at light loads all turns are active, resulting in a low spring scale when the ballhead scale is low, while at high loads the larger turns close out, providing a stiffer or higher scale spring for the higher speeds.

Despite the obvious changes which have taken place during the evolution of the ballhead from its crude

beginnings, the qualities which recommend it remain substantially the same it is rugged and simple mechanically, has a useful output at a reasonably high power level, is relatively insensitive to temperature changes, and is simply adjusted as to speed setting. For these and other reasons this type of speed sensing device is widely used in our governors in conjunction with a servomotor to provide a high power level output.

Various types of servomotors are worthy of consideration. In the case of hydraulic turbine gates it is obvious that the energy required for their operation makes direct control by the speed sensing element impractical. In the early water wheel governors the power required was taken mechanically from the controlled water wheel through clutches selectively operated by the ballhead. As control power requirements became greater and higher speeds of gate movement were made necessary by the demand for better regulation, the practical limit of the simple clutch system was reached. Even for power requirements well within its range, this mechanism has certain disadvantages as a control servo, such as the difficulty of securing and maintaining a small dead band, and, more important the fact that its speed is proportional to prime mover speed regardless of magnitude of speed error. The latter characteristic makes stable operation difficult because of the tendency to overcorrect for small errors. It is desirable to have the speed of servo movement proportional to speed error at least for small errors, and an approximation of this can be secured by intermittent operation of the clutch, decreasing the per cent of time engaged as the unit speed appreaches the proper value. However, this means added complication and more clutch wear.

With electric motors so common, their use as servo units is an obvious possibility. As in the case of the clutch controlled servo, the power required is a vital factor in determining the suitability of electric motors as servos. For water wheel gates where large forces and rapid movements (that is, considerable power) are involved, the size and cost of the electric servo becomes objectionably large. Furthermore, control such a motor, continually started, stopped and reversed, is a serious problem. A further objection is that, the inertia being relatively high, the speed of response is slower than desired.

For relatively low power requirements, or where high speed of response is not essential, the electric servomotor finds entirely satisfactory application; but for our purposes it does not compare favorably with the hydraulic servo.

The hydraulic servo has a great many desirable characteristics to recommend it and comparatively few objectionable features.

Without storage accumulator, the combination of hydraulic servo and pump driven by the controlled prime mover offers a relatively inexpensive, lightweight and compact means of providing power for control. With accumulators, power can be supplied for a short time, say long enough to make one or two full servo strokes, at a very high level without the necessity for increasing pump size (Fig. 19). This is important where pumps may be electric motor driven or where a limited volume of oil makes heating a problem. Control of a hydraulic servo is very readily accomplished with little expenditure of energy by means of a piston type valve. If relative rotation between plunger and sleeve of such a valve is used, frictional opposition to motion is substantially eliminated so that speed sensitive elements of low energy output may actuate directly a valve controlling a fairly large and fast servo. Furthermore it is inherent in such a control that the speed of servo movement decreases as the valve approaches its equilibrium position. The mass of the moving parts is so low compared with the force available, that the response of the hydraulic servo is extremely rapid. If oil is used as the hydraulic fluid, lubrication is inherent. True, the possibility of oil leaks, difficulties with flow at extremely low temperatures, and the necessity for maintaining the oil clean may be considered disadvantages, but they are at most no greater than those inherent in other systems and are for our purpose far outweighed by the advantages outlined above.

Having given consideration to the relative merits of the various governor components discussed above and the accumulated experience of ourselves and others, we have chosen to build our governors around the hydraulic servo of the reciprocating piston type and the centrifugal ballhead actuating a piston type pilot valve. Other types of hydraulic servo such as the vane type unit capable of movement of less than one revolution, and the hydraulic motor capable of contrinuous rotation have found special application, but the simplicity and dependability of the reciprocating piston have thus far not been surpassed for general use. Three types of reciprocating piston servos are used.

These are the double acting piston, requiring a pilot valve having two control lands simultaneously regulating flow of oil to and from opposite sides of the piston and connected to its load by a relatively small diameter piston rod projecting through a seal at one end (Fig. 19); the single acting spring loaded piston, in which hydraulic force under control of a single land pilot valve overcomes spring and external load in one direction and the spring discharges the oil from the cylinder and overcomes external load in the other direction (Fig. 20); and the differential piston, in which in effect the piston rod becomes large enough to reduce the effective area on its side of the piston to one-half that of the other side. In this case oil at supply pressure is maintained in the small area end of the cylinder and oil controlled by a single land valve is caused to flow in and out of the other side (Fig. 21). In all cases the design is such that substantially identical forces are available in both directions of motion. All three of the above servo types are used in Woodward governors, the choice being determined by the circumstances surrounding each governor requirement.

The basic principle of operation of the centrifugal ballhead and the general form in which it is used in most Woodward governors has been described above (Fig. 15). However, ballheads appear in a number of variations, the design being dictated in detail by the specific requirement. The same basic principles apply to all. The servomotor is so simple in principle as to require no further description than has already been given.

Let us now consider the simplest form of hydraulic governor in which the ballhead and pilot valve described control a simple reciprocating piston servo. An adequate oil supply is assumed to be available. For the sake of discussion let us assume that the governor is to control a diesel engine. The ballhead is driven at a speed proportional to that of the engine and the serve is connected to operate the fuel racks (Fig. 22). It should be noted that the simple combination of ballhead and directly connected pilot valve has only one equilibrium position the position in which the valve is closed, neither admitting oil to nor discharging oil from the servo cylinder. For a given setting of the top end of the speeder spring, the ballhead and valve will take this position at only one speed; in other words, such a speed sensitive device is inherently isochronous. Unfortunately, such a system is also inherently unstable. This is because the engine speed does not instantly assume a value proportional to the rack position due to the inertia of the rotating mass. Therefore, if the engine is below the governor speed setting, the pilot valve is positioned to move the servo to increase the fuel. By the time the speed has increased to the setting so that the valve is centered and the servo stopped, the fuel has already been increased too much and the engine continues to speed up. This opens the pilot valve the other way and beings to decrease fuel. As before, when the speed gets to the right value, the fuel control has travelled too far, the engine underspeeds, and the whole cycle continues to repeat. Some means for stabilizing such a system must obviously be added to the two components we have described to secure a satisfactory governor.

The simplest method of securing stability in the system described is to add means which will provide Speed Droop in the governor which in turn results in Regulation in the governed system. The distinction between these terms will appear as we proceed.

For some obscure reason, the understanding of speed droop seems to give difficulty out of all proportion to its complexity. It is nothing more than the governor characteristic which requires a decrease in speed to produce an increase in throttle or gate. Since an increase in throttle or gate is required if the prime mover is to carry more load, it follows that increased load means decreased speed. Why the addition of speed droop should stabilize an otherwise unstable system is readily seen from a consideration of the differential equations of the two systems. However, since this explanation is not universally satisfactory, let us consider an analogy within the experience of all of us.

Assume that the machine which is to be controlled is an ordinary automobile with foot throttle. The driver of the car is to function as servo in operating this throttle. For speed indication we will use the speedometer, slightly modified; that is, we will replace the speedometer needle with a solid disc, half red and half green, with the dividing line where the needle pointer formerly was. On the face of the speedometer we will place an opaque disc having a very narrow slot near the edge through which we can see the indicating disc. This slot may be positioned at will as an adjustable speed setting, but we will assume it to be at the location corresponding to 50 mph, the speed which we wish to maintain. The driver is to attempt to maintain this speed by reacting to the speed indication as follows: whenever the slot appears red, indicating an overspeed, the throttle is to be completely closed; if green, completely opened. It is assumed that someone else will watch the road. Suppose our car to be rolling along at equilibrium at 50 mph and to come to a slight upgrade. The speed will drop, the slot will show green, and the driver will tramp the throttle to the floor. The car will then speed up, the indication will turn quickly from green to red as it goes through 50 mph, and the driver, his reaction time being what it is, will, an instant later and having overshot, completely close the throttle. The speed will then drop quickly through 50 mph and the cycle will be repeated indefinitely. This is a system without "speeddroop".

Now let us assume that the slot is increased in width so that its lower edge is at 45 mph and its higher edge at 55 mph. Our instructions to the driver are that when the slot shows all red, which will occur at 55 mph, the throttle is to be completely closed; when all green, or at 45 mph, it is to be fully open. For intermediate speeds or ratios of green to total slot width, the throttle must take a proportional position. With such an arrangement, starting again from equilibrium, the slight decrease in speed caused by the upgrade does not change the indication to all green, but only increases the amount of green which results in a small increase in fuel, not a complete throttle opening. The car will take the grade at a slightly reduced but stable speed, and the throttle will not be completely opened unless the grade is sufficiently severe to require full power at 45 mph. Such a system has speed droop. In effect it gives the driver (or servo) a chance to start correction before the desired operating point is passed.

Your attention is called to the fact that the simple mechanical flyball governor in which the throttle is operated directly by the ballhead has speed droop inherently, since the only way the throttle opening can be increased, for instance, is by the flyweights moving inwardly, which requires a decrease in speed.

Speed droop in a simple governor such as our present example can be provided by a mechanical interconnection between servo (and therefore throttle) movement and governor speed setting such that as fuel is increased the speed setting is decreased. Such a device may consist simply of a lever of suitable ratio between servo and speeder spring (Fig. 23). The equilibrium

relationship between speed setting and servo position for such a system may be presented by a line sloping or "drooping" downward to indicate decreased speed setting with movement of the servo in the "increase fuel" direction. For each position of the manual speed adjustment there will correspond a sloping line such as that shown in Fig. 24. In this example, the speed adjustment is assumed to be such that if the servo is at its extreme position in the "increase fuel" direction, the governor speed setting is 1000 rpm. As the servo is moved to the opposite limit of its motion, the speed setting is increased to 1040 rpm. The speed droop is usually expressed as the per cent change in speed setting for full servo stroke referred to the speed setting at the maximum fuel position. In this case the governor is adjusted for 4% speed droop. This characteristic is a function of governor design and adjustment only.

Regulation, however, is dependent not only upon the speed droop setting but also the percentage of governor servo stroke required to move the throttle between no load and rated load positions. It is the steady speed rise resulting from decrease in load from rated value to zero expressed as a percentage of rated speed. If the governor discussed above were so connected to the throttle of its engine that only 50% of the servo travel were required to move the throttle between no load and rated load, the regulation would be 2% although the speed droop is 4%.

Speed droop in a hydraulic governor is not always attained by operation on the speeder spring as described above. It can also be secured by changing the position of the flyweights (and therefore their speed for a given speeder spring setting) required to center the control valve at its equilibrium position. This might be done by using a floating lever connection between speeder rod, servo, and pilot valve (Fig. 25), or by having the bushing movable by the servo (Fig. 26). It is sometimes desirable to have an isolated prime mover run isochronously or perhaps the allowable regulation is not sufficient to privide stability. In such cases, we resort to transient speed droop, or, as it has come to be generally called, compensation. This calls for the introduction of a temporary readjustment of speed setting with servo movement to produce the stabilizing speed droop characteristic, followed by a relatively slow return of speed setting to its original value.

This can be accomplished in a number of ways. Present day hydraulic governors use a variety of schemes, perhaps the most common (Fig. 27) involving a floating lever connecting speeder rod, pilot valve and a receiving piston which is urged by spring force toward an equilibrium position. So long as the receiving piston is in this equilibrium position centering the valve requires that the flyweights be always in the same position. For a fixed speeder spring setting this means that the ballhead must, for final equilibrium, be running always at the same speed. The receiving piston is displaced from its equilibrium position by a flow of oil initiated by movement of a transmitting piston which moves with the throttle—or gate-actuating servo. Thus if there were no leakage from this compensating hydraulic system the receiving piston would move as though rigidly connected to the servo and permanent speed droop as described above would result, the centering springs being ineffective. However, if an adjustable leak in the form of a needle valve is provided between the compensating hydraulic system and an oil sump, the centering spring is permitted to return the receiving piston slowly to its initial position after a disturbance by forcing oil out or drawing oil in through the needle valve as required. As this occurs the speed setting of the governor slowly returns to its original value, although the servo and throttle are permitted to remain at the new, different load position. Such a governor is isochronous although provided with the necessary transient droop for stability. This scheme is used in current water wheel, type IC, and type UG governors.

Another method of producing the same result involves the direct (Fig. 28) application of pressure to the pilot valve, adding to or substracting from the speeder spring force in order to effect a change in speed setting. In the adaptation of this method used in the PG governor the oil actuating the servo is required to deflect a "buffer" piston against a centering spring load which produces a pressure differential across a receiving piston rigidly attached to the pilot valve. A needle valve permits equalization of pressure across the pilot valve receiving piston to restore the initial speed setting. In operation, as oil flows to the servo, the buffer piston is moved against the force of its centering spring, resulting in a higher pressure on the lower side of the receiving piston which produces an upward force on the pilot valve. This in effect decreases the force which the flyweights must balance, resulting in centering of the pilot valve at a lower speed which provides speed droop. As the displaced oil is permitted to leak through the needle valve, the buffer piston returns to its equilibrium position, the differential pressure disappears, and the speed setting reverts to its original value. This method, although requiring a more expensive pilot valve construction, has several operating advantages, not the least of which is that exact return of the buffer piston to center is not essential to exact return of the system to normal speed, equalization of pressure only being required.

It is obviously possible to combine temporary and permanent droop in a single governor, and this is done is many standard units where parallel operation requiring proper division of load is anticipated.

Other methods of securing stability are known and have been tried. Among these is the so-called "derivative" control. This means a device sensitive to the first (or higher) time derivative or rate of change of the controlled variable. In the case of a speed control a first derivative system means simply a device sensitive not only to speed, but sensitive also to the rate of change (first derivative) of speed which is acceleration. Acceleration sensitive governors were used long before they were called derivative controls, and we have made an extensive mathematical analysis of

them in various applications. Acceleration sensitivity alone is unsatisfactory, since it will not accurately maintain speed, although its corrective actions are in the directions to tend to do so The minimum requirement in an isochronous control so stabilized is therefore a speed sensitive element and an acceleration sensitive element. Properly designed, it is possible to secure stability in such a mechanical device, but we have not been favorably inclined toward it for the following reasons.

If wander over a narrow band is to be avoided, the stabilizing effect of the acceleration sensitive element must be present for very small accelerations or decelerations. This is very difficult to accomplish mechanically without the use of objectionably large mass or very expensive and delicate construction. Furthermore, mathematical analysis, confirmed by tests, reveals that with rare exceptions, use of the derivative control can accomplish nothing in the way of regulation about an on-speed condition that cannot be secured as well and at much lower cost by proper design of the compensating systems described above. If the disturbance consists of a large change in speed setting rather than a change in load the acceleration sensitive element offers some advantage, but even here it is doubtful if the improvement obtained justifies the complication and cost involved.

What has been said above applies to mechnical-hydraulic governors; it does not preclude the possibility of accelerometric stabilization in an electric governor, and we have in fact such a control system in process of development. However, as implied earlier, other problems present themselves, and the field of application of such a governor is determind by many considerations, including that of economics.

LOAD-SENSITIVE GOVERNOR

Another approach to the improvement of governor performance upon load change requires that the governor be made directly responsive to load changes as well as to speed, with the purpose of securing a corrective change in throttle position before the speed change, which is necessary to make the speed governor fuction, occurs. This can be made quite effective in some applications if properly engineered and designed, and a governor of this type has been developed by Woodward. The scheme which we have used in this design positions the throttle proportionately to the load change by means of an extremely fast hydraulic servo, leaving the speed governor only the job of trimming or correcting for relatively small errors in throttle position. It has the unique advantage of being a very good speed governor even if the load sensing function were to become completely inoperative.

However, load-sensing has certain limitations, which should be noted. For instance, the speed of response and sensitivity of a good speed governor is such that any load responsive device must be extremely fast to gain significant time advantage. This includes, of course, the load-sensing means which in general would not be considered a part of the governor proper. Because of the practical difficulty and cost of sensing mechanical loads (although this is by no means impossible) applications to date have been limited to generator drives. Careful consideration must be given to securing extremely high rates of response in obtaining the load signal which is supplied to the governor in order that the advantages of the high load-governor speed (0.01 sec. full stroke) may not be lost.

If such speeds of response are maintained, the limiting factor in transient performance is the response of the engine itself. The less the total lag in the engine response, the smaller the speed deviation for a given disturbance. However, although increased lag means increased speed deviation for the load-governor, it also requires greater compensation for stability and therefore increased speed deviation for the usual speed-governor also; therefore, even for high lag systems appreciable improvement can be made with a properly designed load-sensing speed governor although not as spectacular as that obtainable on an engine with small lag. In the case of turbo-supercharged engines an additional limitation is placed on the improvement obtainable with the loadsensing governor. Because the air available at low loads, and therefore low turbocharger speeds, is inadequate to burn full load fuel, the rapid movement of throttle to the full load position by the load-sensing governor does not produce the desired result of full load torque. Thus on large load addition performance would not be expected to be substantially different from that obtained with the speed governor alone, which, as has been noted earlier, is quite capable of increasing fuel faster than such an engine can accept it.

PARALLEL OPERATION OF ALTERNATORS

Despite the fact that alternating current generators, commonly called alternators, with their driving prime movers and controlling governors, have been in widespread use in parallel operation for many years, there is a general lack of understanding of such systems even on the part of many operators and engineers directly concerneed. As a result we are frequently called upon to correct troubles which not only do not originate in the governor but about which the governor can do little or nothing.

A simple engine driven alternator supplying an isolated non-synchronous load is relatively easy to understand; but parallel two synchronous units and tremedous changes take place. Governor knobs that adjusted speed now affect load division, mechanical oscillations of the throttle and periodic exchanges of load between alternators arise, and units that performed beautifully alone kick up such a fuss that they won't stay on the line.

Paralleling obviously does not always produce such violent results, but the potential for such behaviour in some degree always exists in parallel operation, and this degree is greatly influence by the characteristics of the controlled unit. For a better understanding of these characteristics we will turn our attention to the alternating current generator.

Speed Control of a Prime Mover

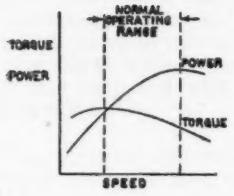


FIG.1

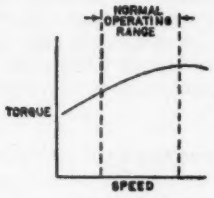


FIG.2

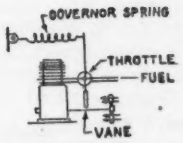


FIG.7

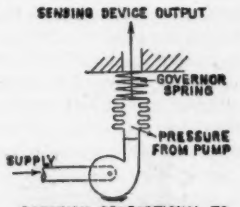


FIG.8

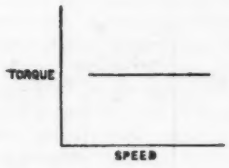


FIG.3

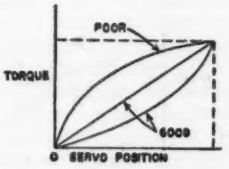


FIG.4

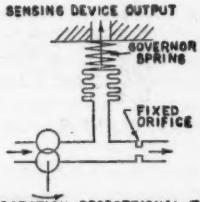


FIG.9

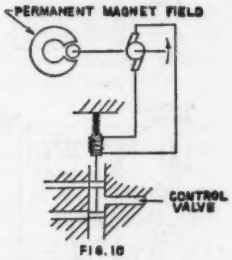


FIG.10

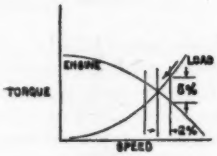


FIG.5

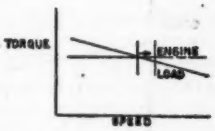


FIG.6

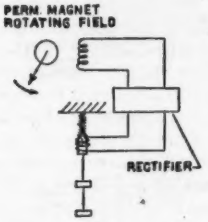


FIG.11

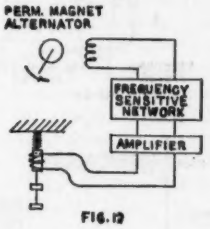


FIG.12

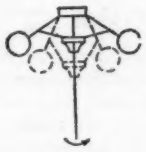


FIG.13

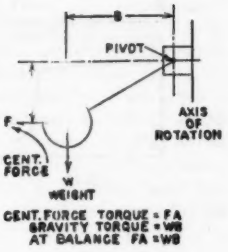


FIG.14

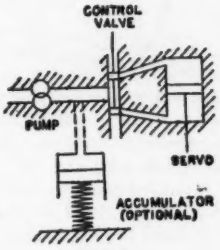


FIG.19

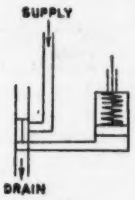


FIG.20

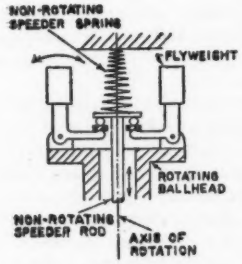


FIG.15

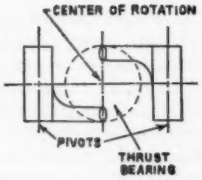


FIG.16

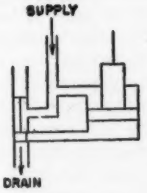


FIG.21

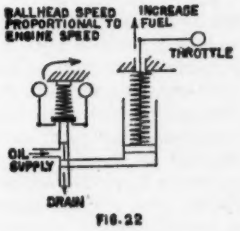


FIG.22

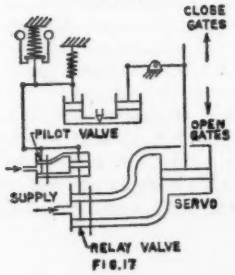


FIG.17

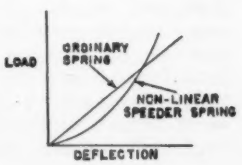


FIG.18

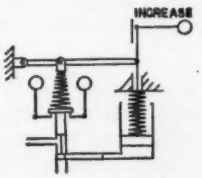


FIG.23

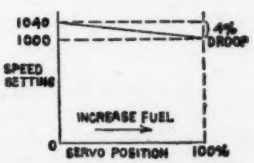


FIG.24

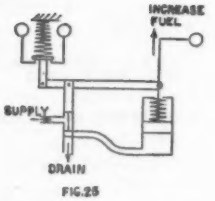


FIG.25

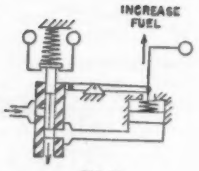


FIG. 26

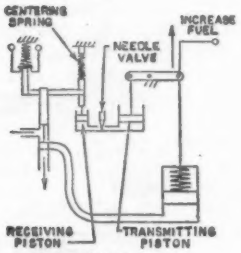


FIG.27

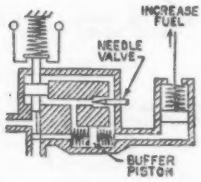


FIG. 28

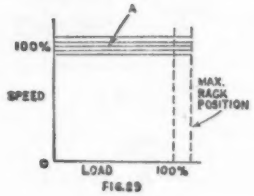


FIG.29

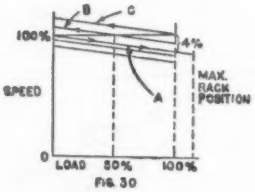


FIG. 30

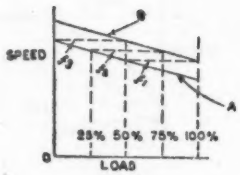


FIG. 31

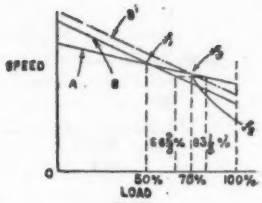


FIG. 32

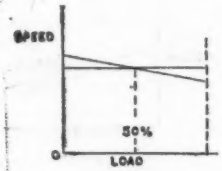


FIG. 33

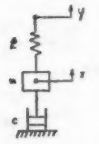


FIG. 34

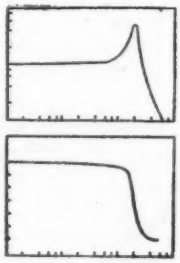


FIG. 35

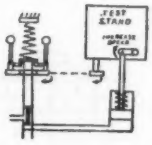


FIG. 36

It is obvious that a simple, nontechnical discussion of alternating current and alternators cannot be very comprehensive but it is hoped that alternator behavior can be made to seem reasonable by means of the over simplified and non-rigorous explanations which will follow.

If an electrical conductor (a wire) is passed sidewise through a magnetic field, a voltage is induced in it, and if a circuit including the conductor is completed, this generated voltage will cause a current to flow. As soon as the current is established, force is required to move the conductor through the field and electric power is developed from the mechanical work done on the conductor during such movement. This is a generator in its simplest form. Conversely, if some outside source of voltage causes a current to flow through the conductor in the magnetic field, force is required to restrain the conductor from moving, and if it is permitted to move against such a restraining force, mechanical work is done. This is the simplest element of a motor.

In an alternator the magnetic field is generally established by causing direct current to flow through coils wound on part of the iron structure of the magnetic path—usually, in the larger machines, on the rotating member or "rotor". Since voltage is generated whether the conductor moves across the field or the field across the conductor, we may choose that the conductor remain stationary while the field rotates. Such an arrangement of nonrotating conductors and iron constitutes the "stator". If alternate north and south magnetic poles pass a conductor in the stator, the direction of the induced voltage, and hence the current, reverses periodically and an alternating current results. The generated voltage in this instance will rise to a maximum value in one direction, decrease through zero to a like maximum in the other direction, again decrease through zero and rise to a maximum in the initial direction. The variation of voltage from one maximum to the next maximum in the same direction constitutes one cycle. In most power generating alternators the machines are so designed and the speeds so chosen that a cycle is completed sixty times per second.

The conductors of the stator, arranged in coils, can be so placed that the maximum voltage in two separate windings will be reached at different times. For instance consider again a rotor having one north and one south pole. A conductor at a given point in the stator will have a 60-cycle voltage generated in it if the rotor makes 60 revolutions per second or 3600 rpm. Another conductor, placed 120° around the stator in the direction of rotation, will reach its maximum at a later time. If we consider one cycle to include 360 electrical degrees, which in this example is produced in 360 mechanical degrees rotation of the rotor, the second voltage will reach its maximum 120° later than the first and may be said to be lagging the first in phase by 120 electrical degrees. If a third conductor is placed an additional 120° around the stator, the three voltages generated will differ in phase by 120°. This is the usual 3-phase voltage relationship. Had the rotating field been made with 4 poles instead of 2, the stator conductors would have been spaced 60 mechanical degrees apart and 60 cycles would have been spaced 60 mechanical degrees apart and 60 cycles would have been produced at 1800 rpm. The electrical phase relationship, however, would have remained 120 electrical degrees. The conductors of such a stator are so connected, that, normally, three terminals result, the voltage between any pair being 120 electrical degrees out of phase with the other two voltages. Since this 3-phase voltage at the terminals results from a rotating magnetic field of constant strength, it is not unreasonable to assume that if an identical 3-phase voltage were connected to the terminals from an external source, a rotating magnetic field of constant strength would result. This is, in fact, the case, and accounts for some of the phenomena with which we will be concerned.

Consider a source of such a voltage having a fixed frequency, unchangeable by anything which we can connect to it; this is the so-called infinite bus. If this source is connected to one side of an open 3-blade switch and our engine-driven alternator so connected to the other side that phase rotation, phase frequency, and voltage are respectively the same, the switch may be closed with no flow of power to or from our alternator. Thus, so far as the engine is concerned, the codition is no different than before the switch was closed so long as the above conditions exist. However the dynamic characteristics of the system which the governor must control have changed greatly, and the effect of this change will be discussed later. The operation just described is performed whenever an alternator is synchronized and put on the line.

If now fuel is increased to the engine, it will try to speed up, but it is no longer free to do so, except momentarily, being forced to run at the average dictated by the infinite bus frequency so long as the alternator synchronizing torque is not exceeded. This torque results from the interaction between the rotating field established by connection of the alternator stator to the bus (the synchronous field) and the DC field of the alternator rotor. So long as they are in phase (that is, lined up) no torque or turning effort exists; if the rotor field leads in phase a restraining torque is exerted, work has to be done by the engine and power flows to the bus; if the rotor lags a driving torque results, power is taken from the bus and the alternator is said to be "motored". Thus there is a torque proportional to the angle between rotor field and synchronous field and in the direction tending to decrease that angle to zero. The effect of this is as though the engine were connected to its load through a torsion spring. Since the speed changes only a very small percentage, the power fed to the bus or taken from it is also proportional to the angle by which the rotor leads or lags the rotating AC field; that is the amount the "torsion spring" is wound up, one way or the other.

If two alternators are paralleled a synchronizing torque exists between them whether connected to a load or not because they are connected to each other, in effect, by the "torsion springs." In this case,

however speed changes are possible if there is no isochronous (constant speed) governor in the system, as will be shown later.

This "wind-up" of the spring in the comparison above is equivalent to displacement of the alternator DC field to lead the synchronous field set up by the line frequency in the actual case. The analogy also holds for the condition of lagging DC field in which power is taken from the line to "motor" the alternator; and for all intermediate conditions including the "linedup" configuration of no power transmitted, which is desired for putting the unit on or off the line. We will now consider the effect of this synchronizing torque on the steady-state parallel operation of governed engine-alternator units.

One of the primary considerations in the parallel operation of power generating machines is the proper division of load. As implied earlier, this can be accomplished by providing a governor with speed droop which results in governed system regulation.

This relationship of regulation to load division is perhaps best explained by reference to the speed vs. load characteristic curves for the governor-engine combinations involved, bearing in mind that the units, when paralleled through alternators, are compelled to run at the same or proportional speeds. For simplicity we will refer to the normal speed as 100% speed and full load a 100% load, and will assume diesel engines as prime movers.

In the controlled system we will be concerned with three types of governor operation: isochronous, speed droop and load limit (which is fixed throttle operation when in control). These we represent, as suggested, as plots of engine load (assumed proportional to servo or rack position) versus speed.

The isochronous governor performance may be represented by a family of horizontal lines, each line representing a speed setting. If set to maintain the speed represented by line A, in Figure 29, and connected to an increasing nonsynchronous load, the speed would vary as follows. Beginning with no load, the speed will remain constant at A as load is increased until the capacity of the engine is reached, and then as load is further increased, speed will drop, the rack remaining at its maximum position.

The speed droop governor has a similar family of curves, but slanted as shown; the greater the regulation, the greater the slope (Fig. 30). Similarly loaded the speed will drop down the regulation line until maximum rack is reached and continue to drop at maximum rack as load is further increased. Suppose now that load limit is set at 50% in either case. The behavior is the same as before except that now the speed drops from the setting line when load limit torque (50%) is reached instead of when the normal rack limit is encountered. Assume the speed droop governed machine is connected to a synchronous system so large that our engine cannot affect its frequency (an infinite bus). Now we must remember that the speed of the engine is no longer determined by the speed setting, but by the frequency of the bus so long as maximum synchronizing torque is not exceeded. What is accomplished by a change in speed setting in this case is a change in load, not in speed. Referring again to Figure 30, we must have setting A for which the no load speed is equal to bus speed (or frequency) in order to synchronize, Once on the line, if we increase speed setting to B we do not change speed but we pick up approximately 1/2 load. Further increase in setting to line C will fully load the engine. If regulation, as indicated by slope of lines, is 4% and the alternator is disconnected from the bus while carrying 1/2 load, the free engine will go up in steady-state speed 2%. If set on line C, carrying full load and similarly disconnected, the no load speed would be 4% above synchronous or 100% speed.

If the speed droop were zero, that is, if the governor were isochronous, paralleling with an infinite bus would be impractical for the reason that a speed setting above synchronous speed by however small an amount would, since actual speed would be below the setting, call for full rack. Similarly if the setting were even slightly below actual speed, the racks would go to fuel off position. Therefore, governors should not be paralleled isochronously with any system so big that the governed unit cannot affect its speed.

If two identical units having speed droop governors are paralleled with each other and there are no other generating units in the system, the frequency will be determined by the speed settings of the governors, the regulation of the governor engine combination and the total load on the system. Here, as before, it must be remembered that the actual speeds of both units must be identical even though their speed settings may be different. Assume both units to be set on curve A, in Figure 31, and the total load equal to the capacity of one unit, that is 100%. In this case each engine carries 50% and the speed is that of line F_1, being the same for both units. Suppose now that one unit is set at B, the other remaining at A. The speed at which both units can run and carry a total of 100% load is now F2, and the load is un equally divided, 75% to the unit with the higher setting, 25% to the lower. The increase in system speed however is by no means as great as the increase in speed setting of B.

If now, with settings A and B, the load is increased to 150%, the regulation being the same for both units the additional 50% will be divided equally between them and A will carry again 50% while B carries 100% The speed of the system will again be back to F_1. Further increase, however, will overload B, so the effective capacity of the system is only 150% instead of the 200% it would have if these speed settings to equilibrium; in fact governors are usually so adjusted inorder to secure the most satisfactory recovery from a disturbance. This fact makes the difference between paralleled and nonsynchronous operation much less obvious because the observer sees only the oscillations resulting from a disturbence, and not the reason for the oscillations.

Consider now the engine driven alternator paralleled with an "infinite bus." One important difference is at once apparent; the speed is no longer determined by the throttle position and load but by the frequency of the bus. The throttle position now determines the poweroutput of the enine and hence the load carried by the alternator. Another and less obvious characteristic is that if a sudden change in throttle position or load occurs an oscillatory transition to the new equilibrium condition results even with the governor inoperative. The explanation for this difference in behavior is to be found in the following.

The effect of the synchronizing torque has been compared to that of connecting the engine to its load through a torsion spring. Insofar as the dynamic characteristics of the system are concerned, this analogy was incomplete in that the mass or inertia of the rotating machine and the damping forces present were neglected. Since the engine and alternator do have mass, this, in combination with the spring action, results in a system capable of oscillation at some natural frequency. If only the spring action of the synchronizing torque and the inertia of the rotating mass were present these oscillations would continue for a long time after a disturbance, as would the swings of a frictionless pendulem suspended in air. Also since the power generated by the alternator is roughly proportional to the position of its rotor relative to the synchronous field, such oscillations would result in power swings of like frequency and proportional magnitude.

In the case of the pendulum mentioned above, the oscillations would die out much more rapidly if it were immeresed in oil. To similarly help damp out the oscillations resulting from a disturbance of the alternator, Amortisseur Windings should be provided in the pole faces of the rotating field structure. These windings act to oppose a change in angle of the rotor relative to the rotating AC field and to damp out the pendulum-like swings by absorbing energy (which is converted to heat) so long as oscillations persist.

Their operation depends upon exactly the same natural phenomena as does the generation of useful power. When the rotor is running synchronously and without oscillation the amortisseur winding bars or conductors do not move relative to the rotating synchronous field. If the instantaneous speed of rotation of the rotor is greater or less than that of the synchronous field, as is the case during oscillation, this relative motion of conductor and magnetic field induces a voltage in the bar which produces a current resulting in a force opposing the oscillation. Because of the resistance of the bars. this circulating current dissipates the pendulum-like energy of the oscillation in heat, resulting in eventual return to a nonoscillating condition of rotation at constant speed.

It may be seen from the foregoing that a fundamental and very important difference exists between paralleled or synchronous operation and isolated or nonsynchronous operation. In the synchronous case the existence of the synchronous field supplied by the bust results in synchronizing and (with amortisseur windings) damping torques which do not exist in the isolated alternator. These torques in combination with the rotating mass result in a "pendulum" system which oscillates or swings after a disturbance, the amplitude of successive swings being decreased or "damped" in proportion to the effectiveness of the damping or amortisseur windings. In the nonsynchronous case there is no synchronizing torque and therefore no "spring" against which the mass of the rotating machine can swing; there is, therefore, no tendency on the part of the unit with fixed throttle to oscillate following a disturbance. This obviously does not mean that an isolated unit cannot oscillate or swing in speed due to other causes such as cyclic variation in developed torque or improper adjustment of the governor. However, oscillations in such cases do not produce the large swings in power which characterize the paralleled case but show up instead as variations in frequency.

The paralleled unit has therefore in common with all such spring-mass systems a natural frequency of oscillation at which it will swing after a disturbance. It is characteristic of such systems that if they are repeatedly pushed at regular intervals of the proper frequency, the amplitude of the resulting load swings may built up to a large value even though the disturbing impulse is small. When the frequency of the disturbing force is such as to produce the maximum amplitude the system is said to be in "resonance". This happens when the forcing impulses occur at the natural frequency of the system. These periodic forcing impulses may come from a regularly varying load, recurring variations in developed torque due, for instance to a weak or misfiring cylinder, or to an unstable auxiliary control such as a voltage regulator or fuel gas pressure control. Regardless of the source, if such disturbances occur at frequencies approaching the natural frequency of the system, load swings will result which may become so large as to make parallel operation impossible. The same unit at the same load on an isolated or nonsynchronous system may be and frequently is perfectly satisfactory in operation, not because the forcing disturbance is any less, but because, lacking the synchronizing torque of parallel operation, there is no resonant condition to produce amplification of the swings. If such a condition exists, the governor can do little to correct it; in fact, in some cases it is aggravated by the governor. Nevertheless, it often happens that the throttle movements produced by the governor in response to the small periodic speed changes that accompany these load exchanges are thought to be the cause rather than the result of the difficulty, as is actually the case. This can be demonstrated quite easily in most cases by blocking the throttles with the load limit or by other means. If the trouble is of the sort described above, the load swings will continue, although some change in magnitude may be observed.

The preferred treatment of this problem is to avoid it by selection of the WK^2 to give a natural frequency

[*Continued on page 129*

Effect of Blade Number on the Torque/Slip Characteristics of a Small Fluid Coupling

By D. FIRTH & I. S. PEARSALL

Notation

Symbol		*Definition*
C	..	Actual linear velocity
C_T	..	Torque coefficient $= \frac{T}{\rho\omega P^2 D5}$
D	..	Diameter
g	..	Gravitation constant (32.2 ft/sec ^{2})
N	..	Speed (rev/min)
p	..	Pressure
Q	..	Circulating flow
R	..	Radii of impeller (see Fig. 11)
R_n	..	Reynolds number
s	..	Slip $= \frac{N_P - N_T}{N_P}$
T	..	Torque
U	..	Velocity of rotation $= \frac{2\pi N}{60} R$
W	..	Circumferential velocity
η	..	Efficiency
λ	..	Loss coefficient $= \Psi/\phi^2$
λR	..	Friction coefficient $= \lambda - \lambda_{SH}$
λSH	..	Shock loss coefficient
ρ	..	Density
σ	..	Slip coefficient $= 1 - (1 - S)(R_1/R_2)^2$
ϕ	..	Velocity coefficient $= W_2/U_2$
Ψ	..	Pressure coefficient $= 2s\sigma$
Ψ_{SH}	..	Pressure shock coefficient $= \lambda s^2 [1 + (R_1/R_2)^2]$
ω	..	Angular velocity

Subscripts

i	..	Inner (radius)
o	..	Outer (radius)
m	..	Mean (radius)
P	..	Pump
T	..	Turbine
1	..	Pump wheel entrance
2	..	Pump wheel exit
3	..	Turbine wheel entrance
4	..	Turbine wheel exit.

During an investigation undertaken by the Fluid Mechanics Division of N.E.L. for the Admiralty, it became necessary to investigate the possibility of using liquids denser than oil in fluid couplings of the Fottinger type. A small hydraulic coupling 4-in. in diameter was therefore made and tested with fluids ranging from thin oil to mercury. At the same time an extensive search of the literature was made for design information. While there is a vast amount of general information available, the particular field of detail design is not well documented. One of the major omissions in the literature is the effect of the number of blades in a coupling on its torque/slip characteristics. Mention is made by Fottinger in a general manner, whilst a more recent paper by Oprecht gives a theoretical treatment.

It was clear therefore that experiments to investigate the effect of blade number would be of value. This was relatively simple to arrange as the small coupling originally designed to study the effect of the specific gravity of the fluid used was easily modified for investigation the effect of blade numbers. The intrinsic interest in the problem was enhanced by the knowledge that couplings are being used on an increasing scale in many branches of industry, and that little research was being done on the subject.

This paper summarizes three separate investigations made over a period of three years whenever staff and equipment were available. The work has suffered from this lack of continuity in staff and equipment and, as a result, correlation between the three series of tests is not always as good as it might be. In addition, these results have been analysed on the basis of Oprecht's theory, which appears to give fairly good agreement.

APPARATUS AND TESTS

The 4-in. diameter coupling as originally designed is shown in Figs. 1, 4(*a*) and 4(*b*) where it can be seen that the blade sections mounted on a ring could be easily removed and replaced by other sets. Tests were made on combinations of blades ranging from 4-36 with both equal and unequal numbers in the two halves of the coupling.

As the work progressed, improvements were made to the apparatus, and three main series of tests can be summarized as follows:—

Series	*To investigate*	*Apparatus*
1.	The use of various fluids	Electrical generator Supplying electric light bulbs, hand tachometer.
2.	The effect of blade number using oil	Rope brake and coupling uncooled, photocell tachometer.
3.	The effect of blade number using mercury	Leather brake, water cooled, Coupling air cooled. Photocell tachometer.

The rig originally used for Series One tests is shown in Figs. 1 and 3(*a*) The input side of the coupling was driven by a 1½ h.p., 1450 rev/min, 3-phase motor, and the output was absorbed by a d.c. motor arranged to run as a generator. The power from the generator was absorbed by a number of electric light bulbs in series. Torques were computed from the output of the generator and its speed. Although it was well understood that this method of measuring torque was approximate, it was considered adequate to give an answer in principle to the problem under investigation.

For Series Two tests the load on the coupling was provided by a simple rope brake as shown in Fig. 3(*b*). The driving motor was the same as that previously used. This brake was simple to make and use and seemed relatively sensitive except for torques lower than 0.8l in. One of the problems on the earlier rig had been the measurement of speed differences without affecting the torque readings by the added friction of the hand tachometers used. To overcome this, a system of speed measurement was developed using two electronic counters operated from photocells. Shutters attached respectively to the input and output halves of the coupling interrupted the photocell beams and thus operated the counter. A "Teflon" lip seal made in the laboratory (Fig. 2) provided a low friction seal round the driving member of the coupling casing. This seal had to be renewed during the tests because of bellmouthing. A more accurate alignment of the two parts might eliminate this effect. This rig was still not sensitive enough to discriminate below 10-per-cent slip, and a new series of experiments (Series Three) was therefore begun using mercury as the working fluid. Because of the higher torques registered, this enabled the range of torques from 0 to 10-per-cent slip to be more thoroughly investigated.

For the Series Three tests using mercury, the apparatus had to be altered in several ways because of the higher powers involved. The final arrangement is shown Fig. 3(*c*), where it is seen that the friction brake was fitted with a water supply system to keep it cool and a ducted fan was arranged to keep the temperature of the coupling within rasonable limits during the course of the experiments, and also to cut down the cooling time between experiments. The use of mercury aggravated the sealing troubles previously mentioned, and after a number of palliative measures had been tried unsuccessfully, the coupling was redesigned, as shown in Fig. 2, to incorporate a Crane mechanical seal Type 109, which as a carbon face and a *NI-resist* seat. It was found that, provided both faces were clean when fitted and the bearing pressure large enough, a good working seal was obtained.

RESULTS

Series One (*The use of fluids other than oil*)

The liquids tried in the coupling were a thin oil mixture, water, carbon tetrachloride, acetylene tetrabromide and mercury. With all these liquids the coupling worked satisfactorily, although the acetylene tetrabromide caused some corrosion. Results of the tests are shown in Fig. 5 plotted on a basis of torque divided by specific gravity against slip. Although these curves are roughly of the same family they do not lie on the same curve. This, no doubt, reflects the limited accuracy of the torque measuring apparatus and, possibly the effects of viscosity. The divergence at the top end of the curves might also be caused by the thermal consequences on the various high rates of slip. It may reasonably be concluded, therefore, that the torque varied directly with the specific gravity of the fluid used.

Because for a given duty the diameter of a coupling varies inversely as the fifth root of the specific gravity of the fluid used, an increase in specific gravity, although reducing the general dimension of a coupling, does not greatly affect the total filled weight of the unit. The use of higher density fluids might also increase the manufacturing costs.

Series Two: (*The effect of blade numbers using oil as the hydraulic medium*).

The working fluid in these tests was Shell Tellus 29 which has a viscosity of 110 centistokes at 70°F.

The relationship between slip and torque for equal blade numbers in the input and output members of the coupling is shown in Fig. 7(*a*), and for unequal blade numbers in Fig. 7(*b*). The insensitivity of the dynamometer and the friction in the coupling bearings caused excessive scatter at low torques and low slips. It will be noticed that the curves for the unequal blades, correspond approximately with the mean number of equal blades (i. e. eight primary and four secondary six on both). Fig. 8 (*a*) shows the torque characteristics plotted against number of blades. This diagram shows clearly the optimum blade number to give maximum torque at any slip. It also shows that if the coupling is to run continuously at low slips, provided a certain minimum number of blades is exceeded, the number of blades is not critical. However, if the coupling is to cover a large range of slips the number of blades for best performance is fairly critical.

An attempt to reduce the effects of temperature which occurred in the coupling, particularly at high slips, was made by starting alternate tests from 100-per-cent slip to get high slip readings with a cool coupling. Each run was kept as short as possible to prevent unnecessary overheating. With the rig as designed, it was impossible to determine the effects of temperature on the torque/slip characteristic.

Series Three: (*The effect of blade numbers using mercury as the hydraulic medium*).

Because of the limitations in the frictional brake it was only possible to measure slips up to 50 per cent. Curves were established for blade number of 8, 16, 24 and 36. These tests were repeated with the coupling half-full of mercury. The results are shown in Fig. 9. On the question of filling, it was found that, provided the torus was completely filled, results were consistent. This represented 80 per cent of the total filled volume of the coupling.

It will be observed that the torque/blade number characteristics for mercury, Fig. 8(*b*), are similar to those for oil, Fig. 8(*a*), but that a larger number of blades is required to secure best operation.

When the coupling was stripped for alteration, it was found that some surface corrosion had taken place on the mildsteel blade sections and that the corrosion reflected the dominant flow patterns in the coupling, Fig. 4 (*b*) It would seem from these patterns that not all the fluid in the working spaces followed a toroidal path, as there is some evidence of separation flow on the downstream faces of the blade, which probably occurs mainly at high slip.

It can be seen from Fig. 10 that the comparison between mercury and oil is not too good. It appears, as a result of the theoretical analysis, that the flow regime using oil is probably different from that using mercury. This is especially so with the larger blade numbers and higher slips, comparison on a Reynolds number basis gives values of R_n 350 times greater for the mercury than the oil.

THEORETICAL CONSIDERATIONS

Oprecht Theory

A semi-empirical analysis given by Oprecht is of special interest as it takes into account the blade number. Applying it to the N. E. L. coupling, in which the blades are straight and no vortex or load ring is fitted, it is assumed that the flow is completely circular in the passages and not affected by interference between the blades. To simplify the problem further it is assumed that the transfer of energy is concentrated on the mean streamline (defined as the centre of the flow patb). Referring to the velocity triangles, see Fig. 11, the torque acquired by the pump is :—

$$T_p = \rho \frac{Q}{g}(R_2 U_2 - R_1 U_4)$$

and the torque given to the turbine is :—

$$T = \rho \frac{Q}{g} (R_2 U_2 - R_1 U_4)$$

hence as the mass flow is the same $T_p = T$

Now $$U_4 = (1-s)\, U_1 = (1-s)\frac{R_1}{R_2} U_2$$

and $$\frac{Q}{g} = \left[W_2 \frac{\pi}{2} R_o^2 \right] \left[1 - \left(\frac{Ri}{R_o} \right)^2 \right]$$

thus the torque transmitted by the coupling is :—

$$T = \left[W_2 \rho R_o^2 \frac{\pi}{2} \right] \left[1 - \left(\frac{Ri}{R_o} \right)^2 \right] \left[1 - (1-s) \left(\frac{R_1}{R_o} \right)^2 \right] R_2 U_2 \quad \ldots\ldots (1)$$

or $$T = f(\Phi, \sigma, \rho, \omega^2, D^5) \quad \ldots\ldots (2)$$

where $\Phi \frac{W_2}{U_2}$ the velocity coefficient .. (3)

$$\sigma = \left[1 - (1-s) \left(\frac{R_1}{R_2} \right)^2 \right] \text{ the slip coeficient} \ldots . (4)$$

$\omega =$ the angular velocity of the pump runner.

Equation (1) approximates to the conventional dimensionless torque coefficient :—

$$C = \frac{T}{\rho \omega_p^2 D^5} \quad \ldots\ldots (5)$$

The characteristics for this coupling are shown is Fig. 10.

The transfer of energy from pump to turbine is not complete, due to "shock" losses and frictional losses. Thus if η is the efficiency :—

$$T_p w_p \eta = T_T w_T$$

$$\eta = 1 - s$$

Now the change of pressure across the pump impeller can be related to the change of velocity head :—

$$p_2 - p_1 = \frac{\rho}{2} \left[U_2^2 - U_1^2 + C_2^2 - C_1^2 + W_1^2 - W_2^2 \right]$$

from the velocity triangle :—

$$W_1^2 = W_2^2 + (s U_1)^2$$

giving $$p_2 - p_1 = \frac{\rho}{2} \left[U_2^2 - U_1^2 + C_2^2 - C_1^2 + (sU_1)^2 \right]$$

similarly $$p_3 - p_4 = \frac{\rho}{2} \left[U_3^2 - U_4^2 + C_3^2 - C_4^2 + (sU_2)^2 \right]$$

Assuming therefore that $C_2 = C_3$ and $C_1 = C_4$

then the difference between the pressure rise in the pump and the pressure drop in the turbine is:—

$$(p_1 - p_2) - (p_3 - p_4)$$

Now the pressure characteristic is taken as :—

$$\Psi = \frac{(p_1 - p_2) - (p_3 - p_4)}{\frac{1}{2} \rho U_2^2}$$

hence

$$\Psi = 2s \left[1 - (1-s) \left(\frac{B_1}{R_2} \right)^2 \right] = 2s\sigma \quad (6)$$

This pressure difference is due to the frictional loss and "shock" loss in the blades, hence:—

$$(p_1-p_2)-(p_3-p_4)=\frac{\lambda\rho}{2}W_2^2$$

from which $$\lambda=\lambda_4^2 \qquad (7)$$

thus the loss coefficient λ is the relationship of the transformation of pressure energy Ψ into velocity energy Φ^2.

The pressure difference across the blades can be further divided up into the pressure difference caused by friction loss, the pressure difference caused by "shock" loss, and the pressure energy converted into kinetic energy. Hence :—

$$\Psi=\Psi_R+\Psi_{SH}+\Phi^2$$

$$\text{or } \lambda=\lambda_R+\lambda_{SH}+1 \qquad (8)$$

hence λ_R as a function of Ψ, should show the influence of blading upon the torque transmitted, independently of other factors.

The pressure losses caused by shock are:—

$$\Delta p_S=\frac{\rho}{2}\left[(sU_1)^2+(sU_2)^2\right]$$

$$\text{or } \Psi_{SH}=s^2\left[1+\left(\frac{R_1}{R_2}\right)^2\right] \qquad (9)$$

Now it has been assumed that the momentum of the working fluid concentrated on the mean streamline has the same moment of momentum as the actual flow:

$$\therefore \frac{Q}{g}R\,C_R=\frac{1}{g}\int d(Qr)\,C_R$$

From this it can be shown that for equal quantities to flow in the top half and bottom half of the coupling, then :

$$R_m=\left[\frac{R_i^2+R_o^2}{2}\right]^{\frac{1}{2}} \qquad (10)$$

Furthermore by considering the energy of the fluid on the mean streamline and the total energy of the fluid:

$$R_1=\left[\frac{R_m^4-R_i^4}{4(R_m-R_m)}\right]^{1/3} \qquad (11)$$

and

$$R_2=\left[\frac{R_o^4-R_m^4}{4(R_o-R_m)}\right]^{1/3} \qquad (12)$$

We have then the following equations :—

$$\Phi=\frac{W_2}{U_2} \qquad (3)$$

$$\sigma=\left[1-(1-s)\left(\frac{R_1}{R_2}\right)^2\right] \qquad (4)$$

$$\Phi=2s\sigma \qquad (6)$$

$$\Psi_{SH}=s^2\left[1+\left(\frac{R_1}{R_2}\right)^2\right] \qquad (9)$$

$$\lambda_R=\frac{\Psi}{\Phi^2}-\frac{\Psi_{SH}}{\Phi^2}-1 \qquad (8)$$

$$T=W_2\rho R_o^2\,\frac{\pi}{2}\left[1-\left(\frac{R_i}{R_o}\right)^2\right]\left[1-(1-s)\left(\frac{R_i}{R_2}\right)^2\right]R_2U_2 \qquad (1)$$

Analysis of Results

In Fig. 6 the pressure coefficients are plotted against slip for this coupling $\left(\frac{R_i}{R_o}=0.385\right)$. It can be seen from these theoretical curves that, provided the coupling is running at low slips, the "shock" loss is only a small proportion of the total loss (5 per cent) which justifies the use of straight blading. However, if the coupling is to run extensively at large slips, there would be justification in having curved blades.

The generalised result by Oprecht's method is shown in Fig. 12, where, by plotting λ_R against $\frac{z}{2\pi}\Phi$, he found that all results of various couplings when plotted form a single curve with very small scattering of points. These couplings ranged from 6 to 80 blades, 800 to 3000 rev/min, and circulation diameter 8¼ in. to 20 in. The shaded area shown in Fig. 12 represents the scatter on these results. The results of N.E.L. coupling (1450 rev/min, circulation diameter 2½ in., blade number from 4 to 36) are also shown, and it is seen that there is more scatter than with Orecht's results. In addition there is discernible a decided effect due to blade number. A third peculiarity is the bending up of the curves at their lower ends (these points correspond to increasing Reynolds number and increasing slip). It seems likely that this effect is due to a change in the flow regime, possibly the advent of secondary flow conditions, as mentioned in section 3.3.

Oprecht suggests, however, that although in full-scale couplings turbulent flow causes a relationship of the form:

$$\lambda_R \backsim \frac{1}{\left(\frac{z}{z\pi}\right)^2\Phi^2}$$

in small model couplings the flow may be laminar and correspond to the form.

$$\lambda_R \propto \frac{1}{\Phi R_n}$$

This is plotted in Fig. 13 for the N. E. L. coupling. It can be seen the agreement is worse than by the other method of plotting these results.

It is interesting to note that with oil results perhaps the best agreement is obtained by plotting λ_R against Φ Fig. 14. In this case, although all the results appear to lie near one line with much less scatter than before, there is still an effect of blade number.

This line can be expressed in the form this coupling using oil:—

$$\lambda_R = 3.97 \times 10^{-8} \left[R_n \right]^{1.73}$$

or more generally for this coupling as Reynolds number = 1900 :—

$$\lambda_R = 0.0186 [\mu]^{1.73}$$

Reynolds number for the Coupling

The Reynolds number is defined as VD/ν where V is velocity, ν—kinematic viscosity and D the diameter. In the case of the coupling, it is not clear what dimension should be taken for D, Oprecht has used the radius of the mean streamline at pump, exit R_2. However, for the purpose of comparison of friction values with those in pipes, it seems better to adopt some unit dependent on the flow passage. It is usual in cases like this to take D=4m, where

$$m = \frac{\text{area of flow passage}}{\text{wetted perimeter}}$$

$$m = \frac{\pi (R_o^2 - R_m^2)}{2\pi (R_o + R_m)}$$

$$= \frac{R_o - R_m}{2}$$

and $= 2(R_o - R_m)$

bance for oil where ν=57 centistokes at running temperature 100°F.

$$R_n = \frac{V D}{\nu} = \frac{W 2}{\nu} (R_o - R_m)$$

$$= \frac{\Phi U_2^2}{\nu} 2(R_o - R_m) \text{ as } \Phi = \frac{W_2}{U_2}$$

$$= \frac{213}{12} \times \frac{2 \times 0.4}{12} \times \frac{929 \times 100}{57} \Phi$$

$$= 1900 \Phi$$

thus the R_n will range from 60 to 950 on this basis.

By Oprecht's method

$$R_n = \frac{W_2 R_2}{\nu} = \Phi \frac{R_2 U_2}{\nu}$$

for oil

$$R_n = \frac{1.44}{12} \times \frac{213}{12} \times \frac{929 \times 100}{57} \Phi = 3450 \Phi$$

for mercury as ν=0.12 centistokes at 100°F

$$R_n = \frac{1.44}{12} \times \frac{213}{12} \times \frac{929 \times 100}{0.12} \Phi = 1.65 \times 10^6 \Phi$$

Thus by Oprecht's figure, R_n oil=115 to 1725

R_n mercury = 55 000 to 800 000.

It is evident, therefore, that the coupling is running in the transition range with oil and the turbulent range with mercury. The differing results between mercury and oil noticed in this report can, therefore, be attributed to the differing states of flow in the coupling.

RELATION OF λ_R TO PIPE FRICTION VALUES

Previous experiments such as Fottinger's [1] tests in a double U-tube, Seeling [3] and Spannhake's [4] experiments and Oprecht's [2] tests on various couplings have suggested that friction values of λ_R of 10 to 100 times the values found in pipes can be attained in a fluid coupling.

The λ_R value in this analysis corresponds to the f term in the pipe friction formula:—

$$h_F = 4f \frac{V^2}{2g} \frac{L}{d}$$

Thus f for pipes would be :—

0. 025 to 0.01 for laminar flow (R_n 500 to 2000).

If follows, therefore, that as λ_R in the coupling is from 5 to 0.8 over the same range of Reynolds numbers, that the friction in the coupling is some 80 to 200 times that in a pipe.

MEAN POSITION OF RADIUS OF VORTEX

In an attempt to explain the variation of curves in Fig. 12 it seemed possible that this might be caused by a shift of the centre of the vortex with increasing slip. As the speed of rotation of the vortex is dependent on the difference in centrifugal force between the pump and the turbine this difference will be greatest when the difference of speed is greatest. Hence the vortex will tend to move out when the slip is high; this will affect the mean radius R_m as this will increase as slip increases.

Now R_1 and R_2 are dependent on R_m, R_i and R_o; thus as R_i and R_o are fixed by the geometry of the

Effect of Blade Number on the Torque/slip Characteristics of a Small Fluid Coupling

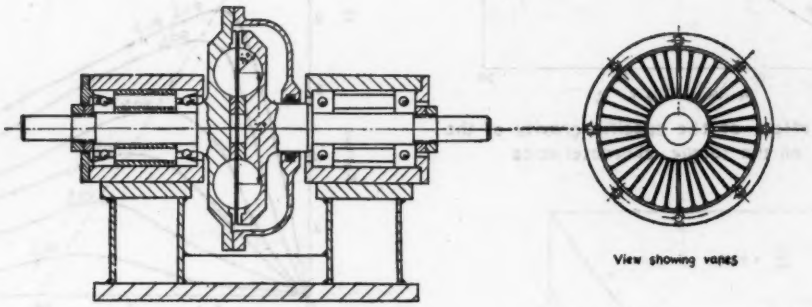


FIG. 1. SECTION THROUGH COUPLING USED IN SERIES ONE TESTS.

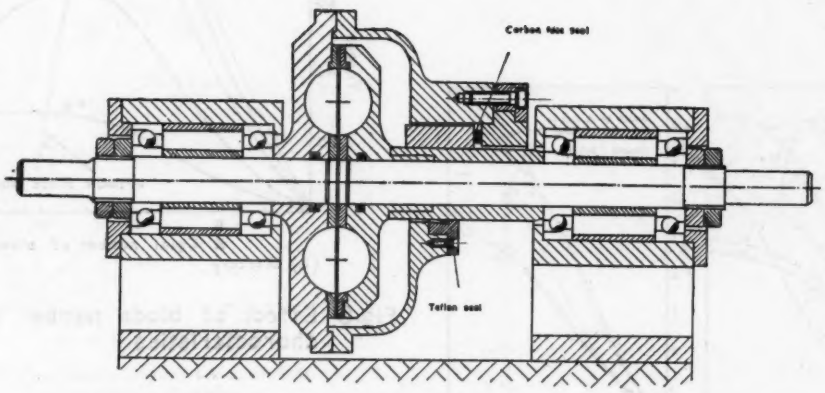


Fig. 2. Section through fluid coupling used in series two and three tests.

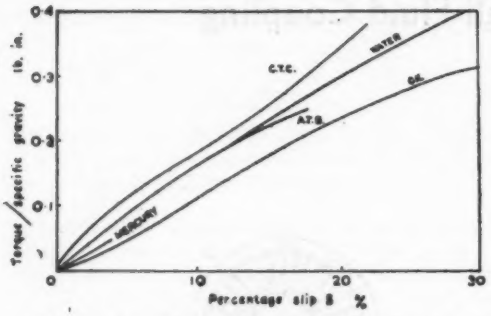


Fig. 5 The effect of the specific gravity of the fluid on the torque characteristics

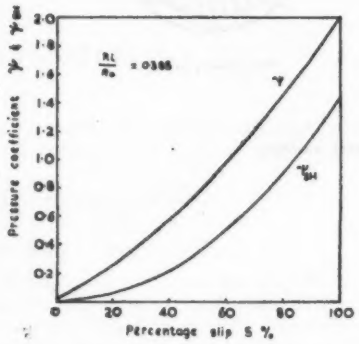


Fig. 6 Effect of slip on pressure losses in blades

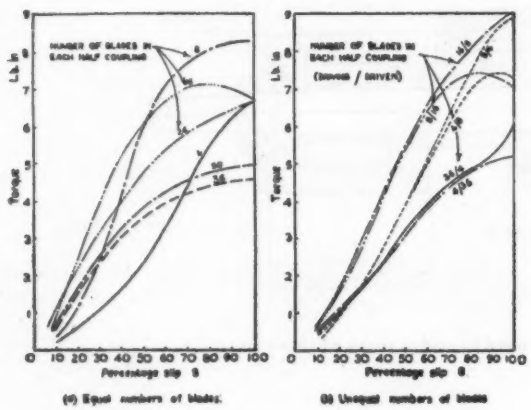
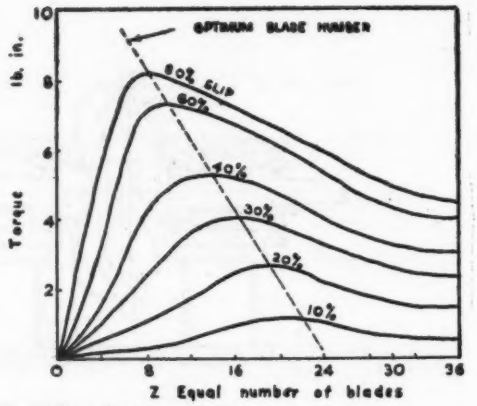
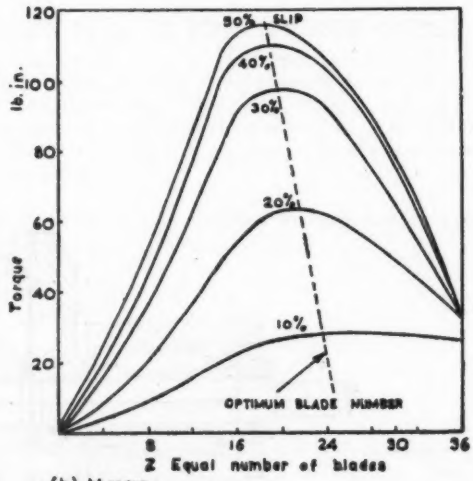


Fig. 7. Torque / slip curves with oil

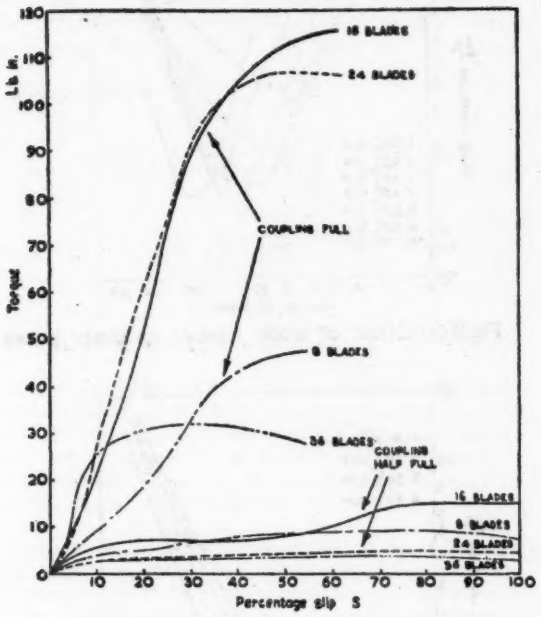


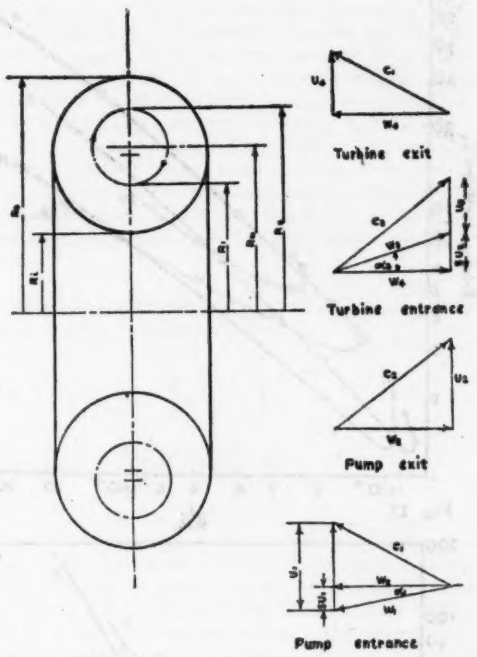
(a) Telius 29 oil

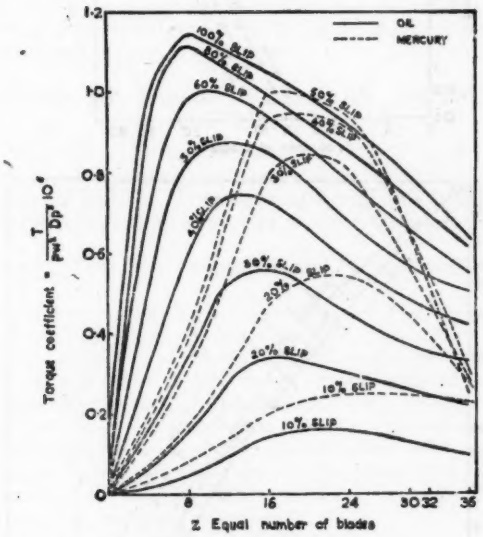


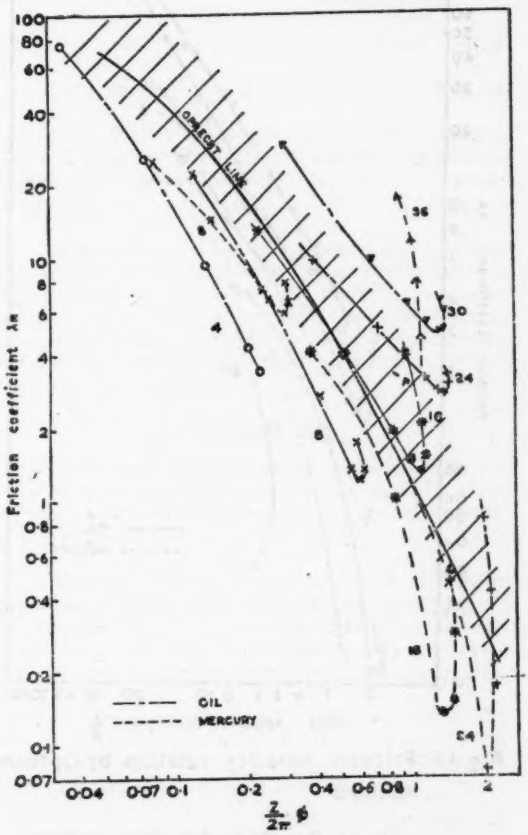
(b) Mercury

Fig. 8 Effect of blade number on torque characteristics









Top: **Fig. 9. Torque slip curves for mercury Equal number of blades**

Bottom: **Fig. 10. Effect of blade number and slipon torque coefficient for oil and mercury**

Right Top:—**Fig. 11. Velocity triangles through fluid coupling**

Right Bottom:—**Fig. 12. Coupling results by Oprecht analysis**

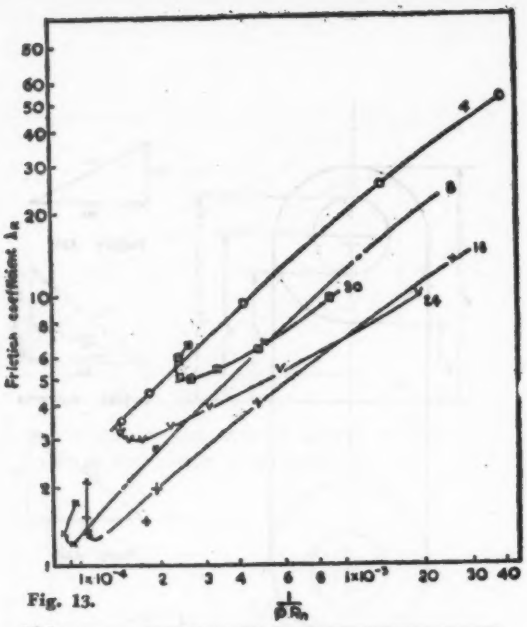


Fig. 13.

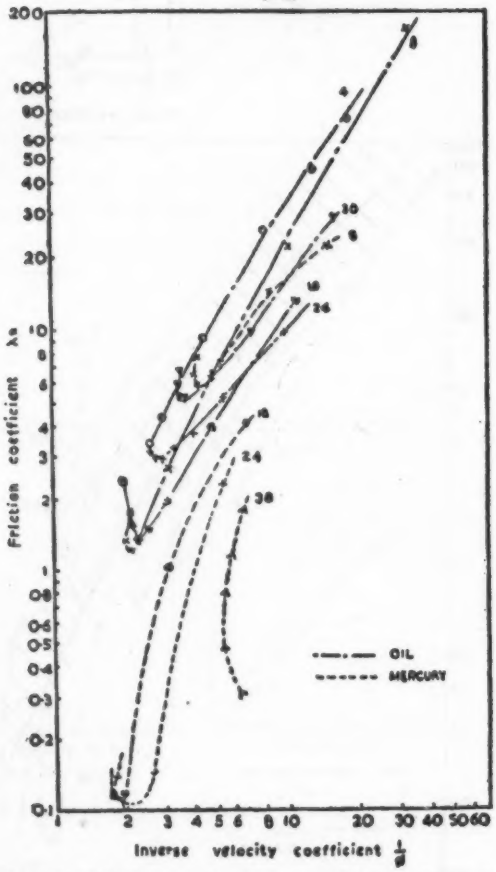


Fig. 14 Friction velocity relation by Oprecht method

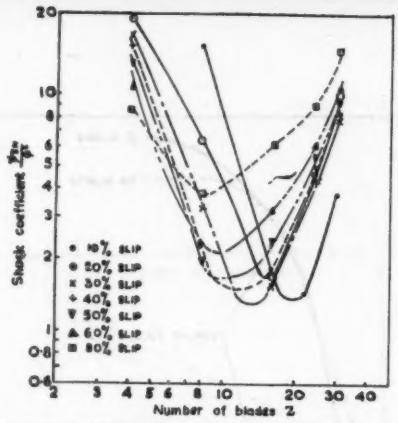
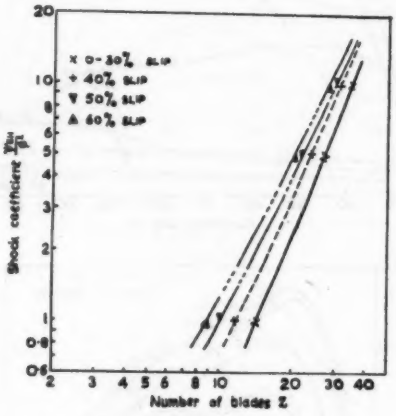


Fig. 15 (a) Effect of blade number on shock losses



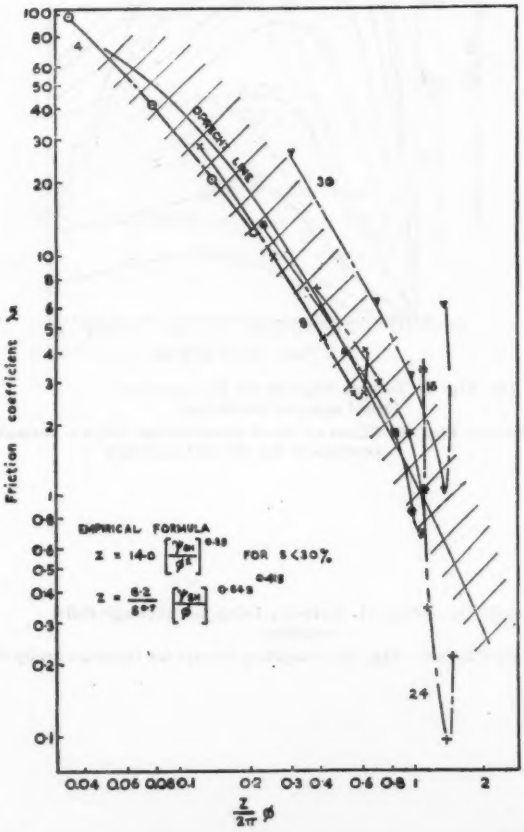


Fig. 13. Oprecht laminar flow condition

Right above—**Fig. 15(b) Empirical correction relating Z, S, & shock losses**

Right bottom—**Fig. 16. Results by Oprecht analysis using an empirical correction**

coupling, then it can be inferred from equation (11) and (12), that the ratio R_1/R_2 will not be affected by a change in R_m.

From equation (3)

$$\Phi = \frac{W_2}{U_2} \propto \frac{T}{U_2^2 R_2} \propto \left[\frac{1}{R_2}\right]^3$$

thus $$\Psi \propto \left[\frac{4(R_0 - R_m)}{R_0^4 - R_m^4}\right] \propto \frac{C_1 - R_m}{C_2 - R_m^4}$$

and if the change in R_m is small then $\Phi \propto \frac{1}{R_m^3}$. Hence if R_m values taken at high slips are too low then calculated values of Φ will be too low. The change in Φ will also affect the λ_z values, and it will cause a reduction in these proportional to Φ^2 2 or R^2_m 6. Therefore at high slips we can say Φ values should be greater and λ_R values will be reduced.

Inspection of Fig. 12, however, shows that although this reasoning may bring the four- and eight-bladed curves nearer to the theoretical line, it will have a reverse effect on the runners with greater numbers of blades. It is therefore clear that the divergence of results from the Oprecht line is caused by some blade effect rather than vortex effect.

EFFECT OF BLADE NUMBER ON PRESSURE COEFFICIENT

When considering the results of Fig. 12 in section 4.2, it was apparent that there was a definite effect of blade number. Investigating further by examining the assumption in each part of the analysis it was evident that the "shock' 'loss $\frac{\Phi_{SH}}{\Phi^2}$ was probably dependent on the blade number as well as on slip and radii of the coupling. (In the basic assumption the fluid is supposed to circulate completely in two planes (x, y); in practice it will have a circumferential component in the plane Z dependent on the number of blades). It is not possible at this stage to develop a theoretical correction for the number of blades. However, based on the assumption that all results should fall on this "Oprecht line", an empirical correction has been determined.

Assuming therefore that all the results fall on this line, working the calculation backwards for the different slips and blade numbers, then the $\frac{\Phi_{SH}}{\Phi^2}$ against blade number has a definite relationship as shown in Fig. 15(*a*). It will be seen that on this basis Φ_{SH} has a relationship to both slip and blade number Z.

It is interesting to note that these curves have two distinct portions joined by a sharp curve. The curved portion, therefore, representing the condition of minimum "shock" loss, and the blade numbers (for each slip) at which this takes place correspond exactly with the blade numbers for greatest torque on Fig. 8. This would appear to be confirmation of the critical effect of blade number, although it must be remembered that the shock coefficient portrayed is calculated on the basis of the Oprecht line and may not be very exact. It does also appear from these two graphs that the number of blades used in production couplings is often greater than necessary, although in scaling up it is more likely that the distance apart of the blades is more important than the actual number of blades.

An empirical relationship for the higher blade numbers (>12) can be expressed:—

$$s < 30\%\ Z = 14.0 \left[\frac{\Psi_{SH}}{\Phi^2}\right]^{0.39}$$

$$s > 30\%\ Z = \frac{6.2}{s^{0.7}} \left[\frac{\Psi_{SH}}{\Phi^2}\right]^{0.64s\ 0.415}$$

When it is remembered that for one coupling Z is constant and s has certain fixed values, these expressions are simpler than they appear at first sight. The curves plotted from these formulae are show in Fig. 15(*b*) and it is seen these are close to the original experimental values.

It is now possible to recalculate the results using these new values of $\frac{\Phi_{SH}}{\Phi^2}$. These are shown in Fig. 16 and it is seen that the scatter of results is greatly reduced. However, the extreme cases of large blade number and large slips still tend to cause wider divergence.

It can be concluded then that use of an empirical formula such as this would need far more results from different couplings to substantiate it before it could be generally used. On the other hand the greater divergence of the results from the N. E. L. coupling may possibly be caused by the greater range of conditions tested on this coupling.

CONCLUSIONS

1. Over a range of fluids which included thin oil mixtures, water, carbon-tetrachloride, acetylene tetrabromide and mercury, the torque in the fluid coupling varied directly with specific gravity.

2. Although the use of mercury as a hydraulic fluid seems to give a satisfactory indication of torque/slip, thus permitting the use of small models, the sealing problems are considerable; particularly in a device of this kind where sealing friction must be a minimum.

3. Because of experimental limitations the results must be considered as tentative. Differences between the oil and mercurry results are almost certainly due to the difference in Reynolds number.

[*Continued on page 135*

P.S.G.
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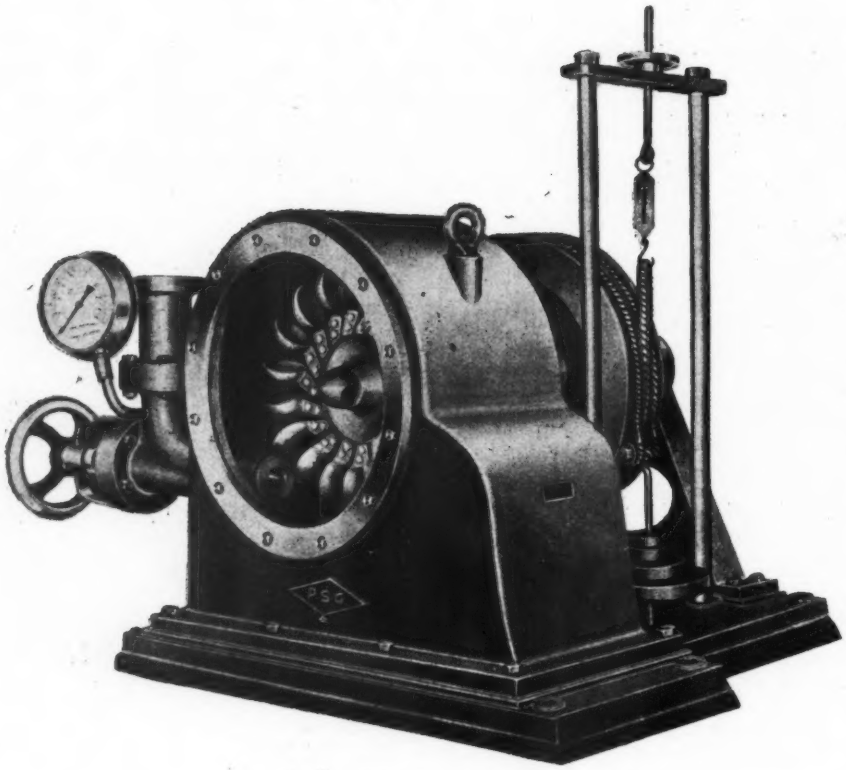

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A Universal Hydraulic Test-Rig of 15 HP and 5 tons capacity

By R. K. K. R. GOVINDARAJAN

Research work of development nature has already assumed great importance in India in all fields of Engineering and Science. Though we can claim that a lot of pioneering research work has been done in India in the flow field of hydraulics, practically no work has been attempted in the field of hydraulic machines. The lack of sufficient empirical data and practical design knowledge makes the development work of hydraulic machines important. Furthermore, it is desirable to do independent work in this field which is rather new to us, with an aim to verify important and available design data and also to establish a more practical and reliable procedure for use in design. Also the experience so gained and built-up will be highly useful for futher advanced research work in this field.

The prototype units and models developed under various research and development schemes need thorough and accurate tests and this necessitates the erection of suitable research equipments and test-rigs. We need many such units for conducting a large number of research schemes facilitating more growth and rapid progress in the field of fluid machinery.

Three development research schemes—one for mixed and axial flow pumps, one for water-reaction turbines of small capacity, and one for venturimeters and orificemeters—conducted in our laboratory required a suitable equipment for testing their units. A universal hydraulic test-rig has been erected providing the following facilities:

1. Testing of pumps of size upto 12″ with a discharge upto 10 cusecs of water at any low head with an absorbing capacity upto 15 hp at speeds ranging from 500 rpm to 3,000 rpm.

2. Testing of turbines of small capacity with a supply upto 10 cusecs of water at any pressure head with an output capacity upto 15 hp at speeds ranging from 500 to 3,000 rpm.

3. Testing of venturimeters, orificemeters, manometers indicators, integrators, recorders or any other flow measuring device with flow rates upto 10 cusecs of water.

GENERAL DESCRIPTION

The general layout is shown in Fig. 1. The layout is prepared for maximum utility and accuracy of measurement required for research and to provide the required convenience in handling the units with the necessary walking space around different units.

The Universal Test-Rig consists essentially of a D.C. dynamometer, a universal test-bed, a supply tank, a weighing tank, a supply pump, piping system and a tilting chute. The D. C. dynamometer is erected in a convenient place and by its side is provided the universal test-bed having easy access to underground tank and supply tank. The weighing tank kept over a weighing platform is near the supply tank. The top tank kept over the supply tank directs water either to the supply tank or to the weighing tank depending upon the position of the tilting chute and the top tank. There is a supply pump on the rear side of the dynamometer with provision to have the intake either from the underground tank or from the supply tank. The delivery line of the pump has provision for mounting venturimeters and orificemeters and it has branches to deliver water to supply tank and weighing tank and also has provision for connection to pump or turbine kept on the universal test bed.

DETAILS OF UNITS

1. D. C. Dynamometer:

This is a direct current dynamometer of the swinging type, capable of working either as a motor or as a generator indicating the shaft horse power. The rated capacity of the dynamometer is 15 hp at speeds varying from 3,000 rpm to 1,200 rpm and proportionately less for speeds from 1200 rpm to 500 rpm. It has double shaft extension and its direction of rotation is reversible. There are suitable loading resistances to absorb electrical energy from the dynamometer when it functions as a generater. The dynamometer is mounted so that the centre of the indicating dial is at about eye level.

2. Universal Test Bed :

This consists of a T-slotted cast iron bed of size 4′-5″ × 3′-6″ with four adjustable cast iron pedestals fixed through the slots for easy and quick mounting of different sizes of test units. An intermediate shaft with a sliding joint and a pair of universal joints, one fixed at each end, facilitates easy connection between dynamometer and test unit eliminating trouble due to misalignment. The construction details for bed and pedestals are given in Fig. 2. The test bed is designed to accommodate units upto base size 4′-6″ × 3′-6″. and shaft centre height upto 2′-0″. The adjustable pedestals provide the vertical movement and the T-slots of the bed provide the horizontal movement. A radial arm provided at the top of each pedestal can cover a radial distance of $2\frac{3}{4}$″ for minor adjustment to accommodate bed bolts without necessitating frequent change in position of pedestal over T-slots.

Supply Tank

The usual way of testing pump, where water is taken from an underground pump and discharged into a measuring device kept on the floor level, suffers from the drawback that pumps cannot be tested at any desired low head, the minimum being limited by the static level difference between the discharge end of pump and the suction level in the pump. A separate supply tank kept at the ground level facilitates the tests on pumps at any desired low head. Mild steel tank was preferred to masonry tank due to easy adaptability for fixing additional parts, minimum floor space and portability. A supply tank of size $10' \times 6' \times 8'$ was fabricated in $\frac{1}{4}''$ thick M. S. sheets. This tank has also provision for use as a Measuring Twin Tank (Fig. 3). The tank has two inside partition walls dividing it into three sections with a central one of size $6' \times 2' \times 8'$ and others of size $6' \times 4' \times 8'$ on either side. The partition walls are provided with circular openings with suitable gun-metal disc valves which can be operated from outside by means of suitable lever arrangement. The valves are so arranged that when one is closed the other is open. The central section has an opening for connection to the suction of the pump. A movable top tank is kept on the supply tank for diversion of water to weighing tank. In case the supply tank is used as Measuring Twin Tank, the top tank is shifted to the centre and discharge from the pump is directed to that end section which is kept open to the central section and the water is circulated for initial running. Water is then diverted to the other end section and discharge measured by volume. The valves are then changed over and the cycle repeated for subsequent tests. The fluctuation of water level in the central section is minimised by a suitable float-operated butterfly-valve. The advantage of the Twin Tank lies in that it does not require any auxiliary pumpset which is needed to replenish water in the case of a supply tank. The diversion of discharge from one measuring section to the other and the change over of disc valves can be simultaneously done by a suitable electromagnetic device. When discharge is measured over a long period and valves are changed over for a number of times, a flow upto 10 cusecs can be conveniently measured with an accuracy of $\pm 1\%$. The supply tank is provided with a suitable overflow device.

Weighing Tank

The weighing tank is of size $6' \times 4' \times 7'$ with a capacity of 5 tons and is made of $\frac{1}{4}''$ thick M. S. sheets. Flow upto 10 cusecs could be measured with an accuracy $\pm 0.5\%$. The top tank is kept adjacent to the weighing tank for diversion of water from supply tank to weighing tank. The weighing tank is also fitted with an overflow device discharging the overflow to the underground tank.

Supply Pump Common Piping System and Tilting Chute

At present, test units of pumps, turbines, venturimeters, orificemeters, etc., are made upto 6″ size and hence only 6″ pipe layout has been used, though upto 12″ size pipe layout could be used. The centrifugal supply pump capable of 1,000 gpm at 50 feet head and 700 gpm at 70 feet head, at 1,420 rpm coupled to a 20 hp squirrel cage motor has been installed with provision in the suction line for its intake either from the underground tank or from the supply tank. The 6″ delivery line of the pump laid just over the floor level has provision to accommodate either venturi meter or orificemeter with a minimum of 15 feet straight upstream length and 7 feet straight downstream length. A sliding joint fitted in the line facilitates easy mounting and dismantling of test units. The overhead extensin of the 6″ line has suitable exits at the required points.

Referring to Fig. 1, the discharge from the pump under test is diverted through a delivery rubber hose to the top tank keeping the gate-valve D closed and the gate-valves A, B and C open. When venturimeter or orificemeter is under test, the 6″ supply pump works gate-valves E, D and C kept open and gate-valve B kept closed. When turbines are tested with the supply pump wsorking, valves E, D and B are kept open and valve C is kept closed.

The tilting chute (Fig. 4) can be fitted to the 6″ overhead line just above the top tank in either of the two positions. The exit section of the chute is $2'\text{-}8'' \times 1\frac{1}{2}''$. By means of a sliding joint and a suitable lever arrangement the chute can be conveniently deflected for discharge measurements.

Pressure Measurement

This is made by means of mercury manometers. The universal manometer developed in the laboratory and presently used for testing various units can be converted to air manometer and used in case of measurement of low pressure. Compound manometers are used for measurement of high pressures.

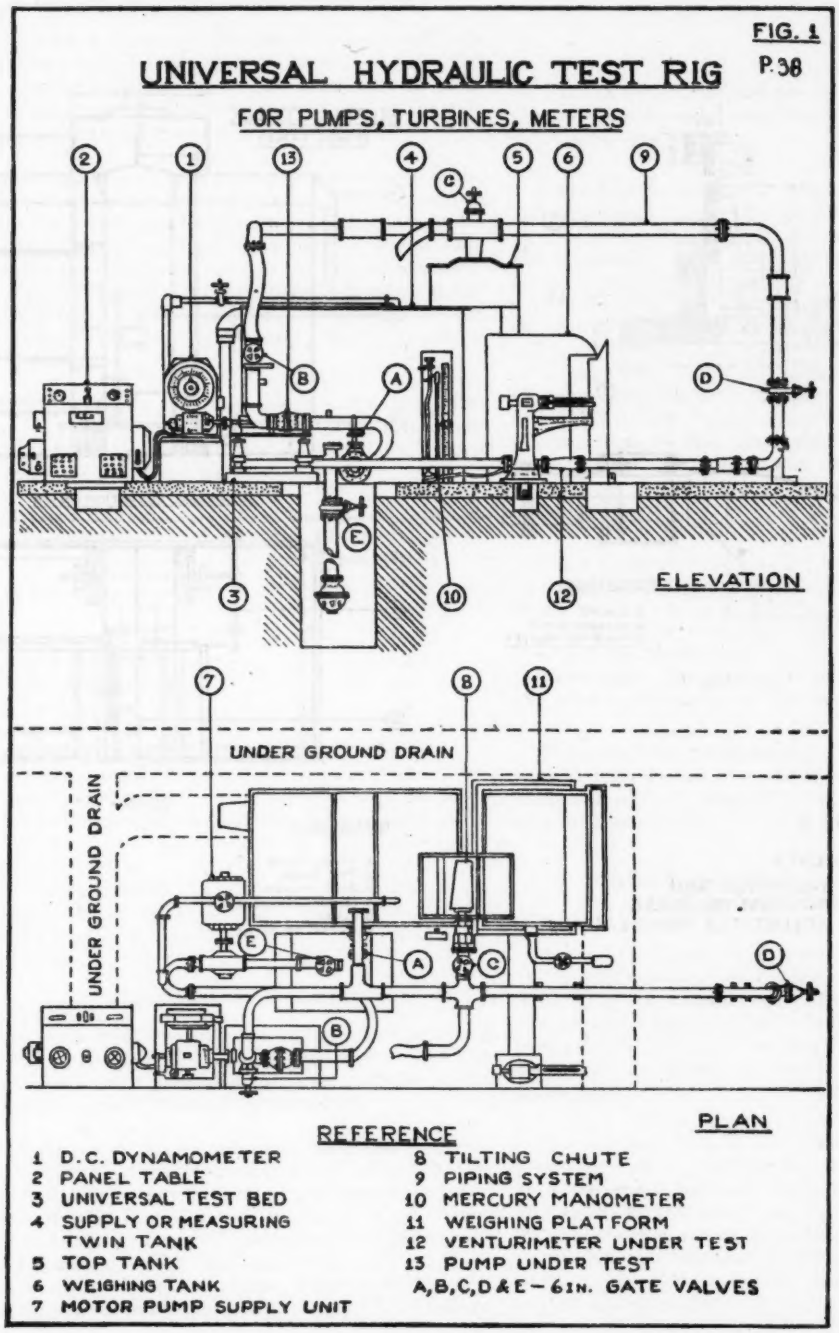
CONCLUSION

It is hoped that the availability of such a versatile test-rig will increase the pace of research work in the field of hydraulic machinery in our laboratory.

Acknowledgement

The Test-Rig was built under the grant of the Council of Scientific & Industrial Research, New Delhi. During the initial stage of work, Sri. P. Radhakrishnan and Sri. V. Mukundan, Senior Research Assistants were helpful in the design and the erection of the rig. The successful completion of the rig is mostly due to the guidance of Sri. G. R. Damodaran and his great help and cooperation rendered through the P. S. G. Industrial Institute, where fabrication of most of the units was done.

A Universal Hydraulic Test-Rig of 15 HP and 5 tons Capacity



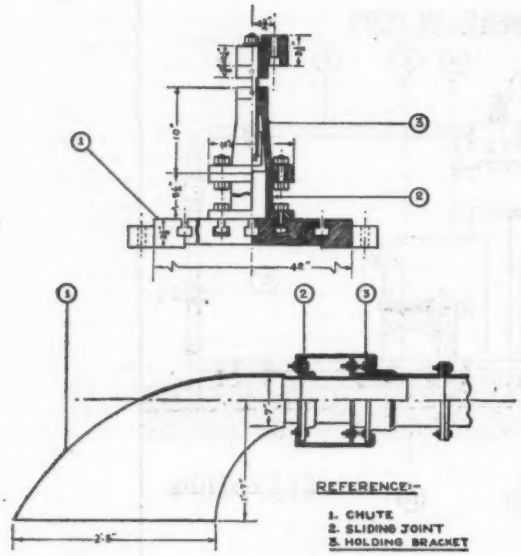
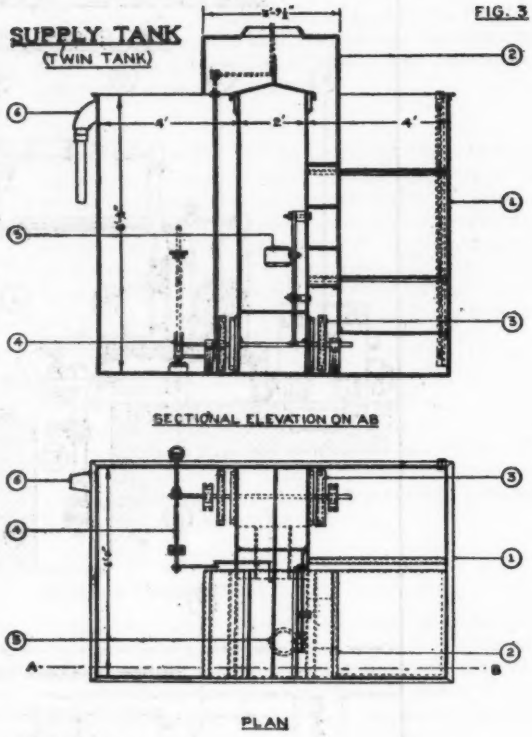


Fig. 4.



Left top :—**Fig. 2.**

REFERENCE :—

1. **T-SLOTTED BED**
2. **BOTTOM PEDESTAL**
3. **ADJUSTABLE PEDESTAL**

On the Design of Machine Foundations

By C. N. NAGARAJ

Special Problems posed by Machine Foundations

The present status of Soil Mechanics is sufficent to deal with foundation problems for static loads. The techniques of soil sampling and the methods of laboratory testing are sufficiently refined to predict the safe bearing capacity of soils under static loads.

But the problems posed by a machine are not simple. The load is not entirely steady and it is repetitional in nature. In addition to its dead weight, there are dynamic loads depending upon the weight of the moving parts, the rate of revolution and the type of machine. The response of soils to repetitional loading is not yet completely understood and therefore the factors that affect the bearing capacity are only partially known.

That different soils have their own natural frequency is now well accepted and the natural frequencies of a few typical soils as determined by Lorenz and Crockett are given below:—

Type of soil	*Nat. Frequency-cps.*
Marsh 10′ thick on sand.	4
Peat.	7.5
Water logged esturine silt.	10
Very light soft clay.	12
Light water-logged sand	15
Medium clay.	15
Wet clay.	19.2
Moist medium sand.	21.8
Dry medium sand.	22
Silt and sand mixed.	23.3
Sand & rubble loosely compacted	23.5
Dry clay.	24.6
Firm surface clay.	25.5
Uniform coarse sand.	26.2
Tightly packed coarse gravel.	30.00
Lime stone.	30.00
Sand stone.	32.00
Granite.	40.00

This table indicates that the natural frequency of soils lies in the range of 4 to 30 c. p. s and that the natural frequency increases with the increase in the allowable bearing pressures on the soil.

If the operating speed of a machine were to correspond with the natural frequency of the soil (or that of the foundation-soil system more precisely) resonance results and this increases the amplitude of vibration. Increased amplitudes of vibration impair the efficiency of the machine and would also cause annoyance to those working near the machine. The soil below such a machine in resonance is brought to a state of failure very much earlier and the consequent replacement of the machine foundations involves loss of man and machine hours.

A machine running at a steady rate is a source of elastic waves to move along all directions in the soil at a frequency corresponding to the operating speed. But a machine like a forge hammer sets the soil into its own natural frequency (or more precisely the natural frequency of the foundation-soil system) and the elastic waves travelling in all the directions in the soil would have this frequency. Objects such as parts of a machinery, light and loose parts of structures, chimneys, etc., whose frequencies are of a similar order are excited causing annoyance and in some cases causing damage also. Annoying effects due to such vibrations are felt miles away from their source depending on the strength of the impulse and the type of soil medium.

Due to repetitional loading caused by a machine, the settlement is much larger than for an equivalent static weight and it is a matter of considerable importance in the design of machine foundations to see that the settlement is a minimum even after a large number of load repetitions in order to maintain the relative alignment of the machine.

These are some of the most important aspects which should not escape the attention of a Foundation Engineer while dealing with the foundations of machines in general.

Present Approach to the Problem of Machine Foundation design

A machine foundation must primarily satisfy all the conditions required for that of a static load. For want of exact information on the factors influencing the bearing capacity of soils under dynamic loads, it is not desirable to load the soil at an intensity greater than half the allowable bearing capacity under a static load.

The concrete bed is made such that there is no possibility of its warping or settling differentially. It is also made perfectly rigid in order that it may absorb the vibrations caused by the rotating machine. To achieve this, empirical methods suggest the use of a certain mass of concrete in proportion to the B. H. P. or the r. p. m. of the machine. Some of the practices in vogue are as follows:—

	Weight of block per BHP developed
(*i*) Gas engines with multi cylinders	1600 lbs
(*ii*) Diesel engines with multi cylinders	1250 lbs
(*iii*) Steam engines with multi cylinders	500 lbs

The weight of concrete for single cylinder engines is generally 40—60% more than the above values in view of the fact that single cylinder engines cannot be as easily balanced as multi-cylinder engines.

Robert's formulae for the weight or volume of foundation concrete are:—

$$W_f = KW\sqrt{N} \quad \text{and}$$

$$V_f = RW\sqrt{N} \quad \text{where}$$

W_f and V_f = the weight and volume of concrete
W = weight of the machine
N = speed, r.p.m.
K and R = constants as given in the table below.

Type of engine	*K*	*R*
(*A*) *Vertical Engines*		
4 cyl. gas engines	0.130	0.000975
3 ,, ,,	0.150	0.001130
2 ,, ,,	0.175	0.001310
4 cyl. diesel engine	0.177	0.00133
(*B*) *Horizontal Engines*		
Single cyl. semi-diesel engine	0.300	0.002250
2 Cyl. -do-	0.240	0.001800
3 Cyl. -do-	0.230	0.001730
4 Cyl. -do	0.225	0.001690

Other empirical methods for the foundations of reciprocating machines are in terms of the bore of the cylinder or in terms of the length of the stroke of the piston. The suggested practice is

Type of Engines	*Width*	*Length*	*Depth*
Vertical Engines	3′—6′ Wider than the base plate	4′—8′	4—5 times the cylinder bore or 3-4 times the length of stroke.
Horizontal-Engines	1′—3′ wider than the base.	Length of foundation base at top should clear the 45° line from the crank shaft.	4-6 times the cylinder bore or 2-3 times the length of stroke.

In all these methods certain amount of importance and stress is laid on the massiveness of the foundation. No consideration is taken of the natural frequency of the foundation-soil system and the possibility of resonance. This is one of the greatest "missing links' in the methods that are in vogue and a more rational approach to the problem of vibration is called for in the design of machine foundations.

Approach to the Problem from the point of view of Vibrations

The natural frequency of the soil at a site can be determined by a simple method. A sudden impulse such as that due to the fall of a spherical weight of about 2 cwts. from a height of 8′—10′ on a thin concrete base embedded in the soil sets the soil into oscillation at its own natural frequency, the energy of the impulse flowing out into the surrounding terrain in the form of waves. This vibration is picked up by a vibrograph suitably set for vertical osillations. The vibrogram which has the trace of the vibration as well as the time base is analysed to arrive at the frequency of oscillation. The natural frequency can also be determined by the resonant frequency method which requires a vibration generator, variable speed motor, electrical equipment, etc. This method is very much more cumbersome than the former one and cannot be recommended for field use.

Having known the natural frequency of the soil, the foundation is so designed that its own natural frequency is of the order of 1/3 of the natural frequency of the soil. This is achieved by properly proportioning the foundation block and supporting it on springs. Crockett has installed dozens of forge-hammers on specially made rubber springs and has prevented the occurence of resonance thus avoiding the (1) loss of power, (2) excessive settlements and (3) large amplitudes of Oscillation. Figure (1) shows a typical hammer mounting and sandwiched type of rubber springs.

In a forced vibration system (as in the case of a steadily operating machine),

$$\frac{X_o}{X_{st}} = \frac{1}{\sqrt{(1-r^2)^2+2\left(\frac{C}{C_c}\cdot r\right)^2}} \quad \text{where}$$

X_o = displacement of amplitude
X_{st} = displacement due to a static load equal to the peak value of the periodically varying load
$r = \frac{W}{W_n}$ = ratio of impressed frequency to natural frequency.
C = damping factor
C_c = critical damping factor of the system.

We can readily observe from this equation that X_o can be made negligibly small by making X_{st} small, i.e., by stiffening the structure. Fig. (2) shows that the amplitude depends upon the relationship between the impressed frequency and the natural frequency, and the amount of damping. If r=1, the amplitudes are

large and for other values of r, the nature of vibrations are as shown in the figure. A significant point that can be noticed from this figure is that the amplitude of vibration is almost equal to the static deflection when the impressed frequency is 0.2 to 0.4 times the natural frequency (or when it is 1.4 times the natural frequency). It is for this reason that the frequency of the foundation-soil system is kept at about 1/3 the natural frequency of the soil, particularly for forge hammers.

Natural Frequency of Foundation-Soil Systems

It has been established by Eastwood that the natural frequency falls with the increase in dead load acting. It is therefore obvious that the natural frequency of the soil at a site is different from the natural frequency of the foundation-soil system. A new method has been evolved by Balakrishna Rao and Nagaraj on the density pressure bulb concept (Reference No.1). This method suggests that the weight of the soil participating in the vibrations of a foundation block is contained in a pressure bulb of intensity D lbs/s. ft where D is the density of the soil at the site in lbs/c. ft. The weight of the soil is given by

$$W_s = 0.1729 \times Q^{3/2} \times D^{-\frac{1}{2}}$$

Where D = density of the soil at the site and the
Q = sum of the static weight of the machine, peak dynamic load and the weight of concrete block. This is treated to be acting as a concentrated load at the surface of the soil.

Using this value of W_s, the natural frequency of the foundation-soil system is obtained by substituting in the equation

$$f_n = \frac{1}{2\pi}\sqrt{\frac{Kg}{W_s + W_m}} \text{ where}$$

$K' = \frac{K}{A}$ = modulus of dynamic subgrade reaction, determined by other daynamic methods.

A = area of the foundation block.

W_s = weight of the soil participating in vibrations.

W_m = weight of the machine and the foundation block.

Once this frequency is determined, it can be examined whether the operational speed of the machine is 0.2 to 0.4 times or 1.4 times the natural frequency of the foundation soil system in order to ensure that the amplitudes are low and also to ensure that there is no resonance. As the speed of a given machine is fixed, the above condition is satisfied by varying the weight of the foundation block.

Weight of the Foundation Block

A new method has been evolved at the Indian Institute of Science by Balakrishna Rao (Reference No. 2) by combining the technique of Newcomb and Hool and Kinnie to determine the weight of the foundation block for any given maximum permissible amplitude of vibration of the foundation block. The weight of the foundation block, W_4 is given by

$$W_4 = \frac{1}{1+K}\left\{\frac{r}{b}\left[W_1 + W_2\left(1+\frac{1}{q}\right)\right] - W_3\right\}$$

where

W_1 = Weight of the rotating parts
W_2 = Weight of the reciprocating parts
W_3 = Weight of the engine
l = length of the connecting rod
r = radius of the crank

$q = \frac{l}{r}$

b = amplitude of the resulting harmonic motion of the foundation block which is generally limited to say 0.005°.

k = ratio of the effective weight of the soil participating in the vibrations to the weight of the foundation block. This depends on the nature and properties of the soil such as density, moisture content, etc.

The above equation may be re-written as

$W_4 = \frac{A}{1+K}$ where A is a constant for a given machine. In the improved method suggested by Balakrishna Rao K is determined by successive approximation as mentioned in the following steps :

(*i*) A random value of K is chosen, say 4, and the value of W_4 is found. Let this be M_o. By definition of K, the product of K and M_o gives the weight of soil participating in the vibrations. Let this be M_1,

(*ii*) By using the method of density pressure bulbs suggested by Balakrishna Rao and Nagaraj (as explained above), the weight of soil actively participating in the vibrations is found, treating the static and dynamic loads as concentrated at a point. Let the weight of soil as determined by this method be M_2.

(*iii*) If M_2 is not equal to M_1, it is due to the arbitrary way in which K is chosen. Obtain another value of K, say K_1, by dividing M_2 by M_o.

(*iv*) Using K_1, repeat the operation once again to get $W_4 = \left(\frac{A}{1+K_1}\right)$. If this is $(M_o)_1$, find $(M_1)_1$ as in step 1.

(*v*) The rest of the steps are repeated till a new value of K_2 is got. By a successive repetition—which may be reduced with certain experience—it is seen that K converges to a constant value and this value

is taken for the final evaluation of W_4, the weight of the foundation block for a permissible maximum amplitude. By keeping this amplitude as low as possible, it is possible to practically control the long-term settlement of a machine foundation.

This method has been developed in a rational way and is a distinct improvement on the methods suggested by Newcomb and Hool and Kinnie.

Permissible ranges of accelerations amplitudes and frequencies

In the design of machine foundations, a watchful eye must be kept on the accelerations, amplitudes and frequencies of the resulting vibrations due to the operation of the machine. Certain combinations of these may result in causing annoyance to human beings and may even result in structural damage. As a guidance to the foundation designers, the following ranges are given.

Vibrations which produce accelerations of the order of 0.01g are just perceptible and if they overshoot 0.05g, the effect is annoying to workers. The effect is awfully painful if the accelerations are more than 0.3g Cracks in walls appear at accelerations of 0.5—1.0g and beyond 1.0g, total or partial destruction of buildings will positively result.

Outstanding work has been carried out by Reiher and Meister under laboratory conditions (and later verified in the field) regarding the combinations of amplitudes and frequencies of vibration which affect human beings. Crandel has also given the combinations of these which cause damage to buildings.

The effects of various combinations of amplitudes and frequencies are graphically illustrated in Fig. 3.

The foundation designer should take care to see that the resulting vibrations due to a machine do not become a nuisance to workers nor do they become a source of dangerous vibrations causing structural damage to buildings in the vicinity.

The problem of isolation of vibrations by techniques like spring mounting etc., has been developed to a great refinement by Crockett and it is another specialised aspect of foundation engineering of machines; it is beyond the scope of this paper.

Foundations of Turbo-Machinery with special reference to Hydraulic turbines

The support for the turbines is the most important part in the design of a power house. The foundation for a turbine should be a solid R. C. C. mat of several feet in thickness, separated from the rest of the building foundations to avoid transmission of vibrations. While it should satisfy all the well known conditions for static loads and the factors mentioned earlier with regard to vibrations, the design of turbine foundations should take account of special factors, the likeof which are not commonly encountered in the case of other machines.

In view of the large magnitudes of torsional stresses, the supporting beams must be adequately reinforced to take torsion. In general the allowable stresses in steel and concrete are kept reasonably low and in order to take secondary stresses, it is essential to maintain a certain minimum percentage of reinforcements in all the parts of the sub-structure.

On account of the impracticability of calculating the natural frequency of such large structures which are by no means geometrically symmetrical, the problem of vibrations is eliminated by making the supporting members heavy enough and a certain ratio of dead loads to live load is always maintained. Observations at the world's largest installation show that the mass of concrete is atleast $2\frac{1}{2}$ times the weight of all the rotating parts or $1\frac{1}{2}$ times the total weight of the turbine and generator, this proportion being maintained for each pier and beam.

It is well known that the natural frequency of a structural member, such as a beam, is related to its own deflection under static load. For an elastically supported concentrated load,

$$\text{Frequency (cpm)} = \frac{188}{\sqrt{\text{Static deflection (inches)}}}$$

From this it is seen that certain static deflections must be strictly avoided in order to prevent the resonance between the structural member and the turbine. As an example it may be stated that a beam whose static deflection under peak load is of the order of 0.004" has natural frequency of 3000 c.p.m. If the machine supported by such a beam were to run at a speed of 3000 r. p. m., resonance and the consequent excessive amplitudes of vibration will result. It is a good practice to keep the static deflection as low as possible and at any rate not to exceed 0.02".

There are other causes which give rise to vibration trouble in turbine foundation practice and care must be taken to see that the effect of such vibrations is kept as low as possible.

Den Hartog has illustrated a case where an improper relationship in the number of guide vanes and the number of buckets gave rise to a loud hum which was heard for several miles. Very bad virations may result from an improper relation between the speed, number of guide vanes runner buckets, length of spiral case and the velocity of propagation of water hammer waves.

Operation of turbine wicket gates cause pressure surges which in turn give rise to vibrations in turbines. The frequency of such a pressure surge, F. (c. p. s) is given by

$$F = \frac{V}{4L} \text{ where}$$

V=Velocity of water hammer waves in penstocks (ft/sec.) which may exceed 4200 ft/sec. in actual pipes

L=Length of penstock in ft.

On the Design of Machine Foundation

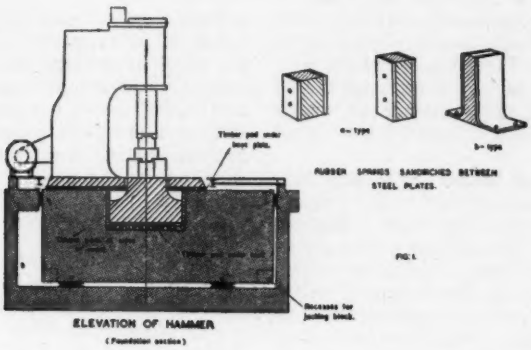


Fig. 1

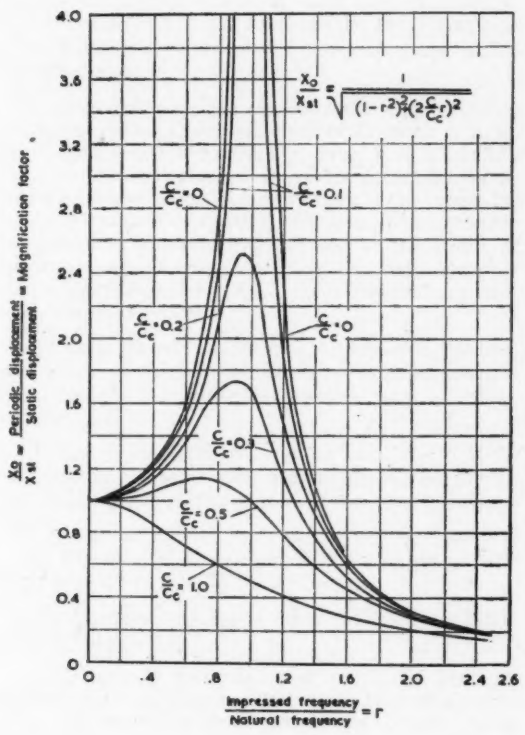


Fig. 2

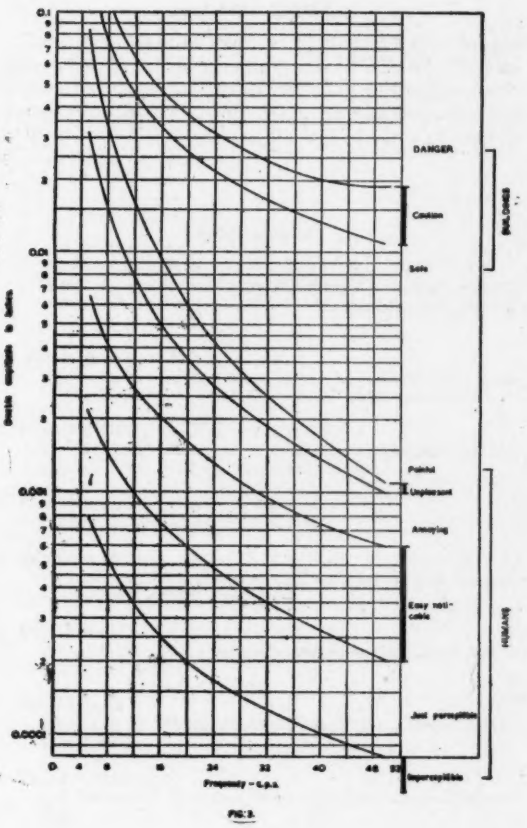


FIG 3.

The effect of these vibrations are rendered almost negligible by the wave reflections and pipe line friction.

Surges in draft tubes are believed to be caused at times due to the presence of air or vapour accumulated around the vertical axis of the draft tube below the runner, when the turbine is running at partial gate openings. This affects the effective head on the turbine and consequently the power out-put of the generator. This also produces changes in pressure in the penstock in addition to water hammer. Frequency of the draft tube surges in c. p. s. is given empiricially by N/216 where N is the speed of the turbine in r. p.m.

It has been reported that the vibrations of blades of a large diameter runner at the Parker Power plant have resulted in the cracking of the blades. This damage is said to be caused by the sudden changes in pressure at the trailing edge of runner blades which result in setting the blades into vibration.

Recommended Practice

As seen from the above, the sources for vibrations are many and the analysis of reach one of them and more so the analysis of their combined effect is extremely complicated, if not impossible. The frequencies of each one of these causes lie in a wide range of 1.5 to 150 c. p. s.

In view of these wide variations, attempts at correlating the mass of foundation concrete with r. p. m. head, BHP, Specific Speed, etc., have not been fully successful. Investigations of several power house installations have been made to arrive at a reasonable empirical constant relating the mass of foundation concrete and the power generated or the mass of concrete and the weight of rotating parts. Even in this respect the mass of concrete to kilowatt capacity varies from 1 to 30.

Therefore, strictly based on the experience of several power houses which have behaved well during the past few decades, the following practice may be safely recommended:

(*i*) The ratio of the volume of concrete in c. ft. surrounding the scroll case and supporting the generator to the capacity of the generator in kilowatts is kept at 2.

(*ii*) The ratio of the weight of the rotating parts to their support is 2 to 2¼.

(*iii*) Accelerations due to any periodic force is limited to 0.005 g to 0.1g.

(*iv*) Structural analysis must take into account forces due to earthquakes and wind. In dealing with these, the resulting force (horizontal or vertical) is taken as the product of the mass of the structural member and the maximum allowable acceleration.

(*v*) On account of the complicated geometry, complex connections and fixities and massive nature of the generator-turbine foundations, it is not possible to calculate its natural frequency. The best that can be done is to simplify the system as much as possible and obtain the lowest natural frequency on the basis of the density-pressure bulb concept explained earlier. This predicted natural frequency must be made far higher than the impressed frequency due to the running of the turbines. This is achieved by suitably altering the total weight of the foundation concrete.

Inspite of the recent researches to rationalise the design of machine foundations in general, it must be concluded that the design of turbine foundations is still based on empiricism. As explained earlier, this is rather inevitable to some extent on account of difficulties of analysis of a huge massive structure. A large scale intensive study of several existing Power Houses with respect to vibrations particularly round the scroll case, draft tube, supporting beams, floors and columns may yield valuable results to arrive at a better approach to this problem.

Acknowledgment

The author thanks Prof. N. S. Govinda Rao for his keen interest in stimulating research in Soil Dynamics at the Institute and for his valuable encouragement. Thanks are also due to Mr. H. A. Balakrishna Rao for helpful discussions and for the use of some of his yet unpublished work.

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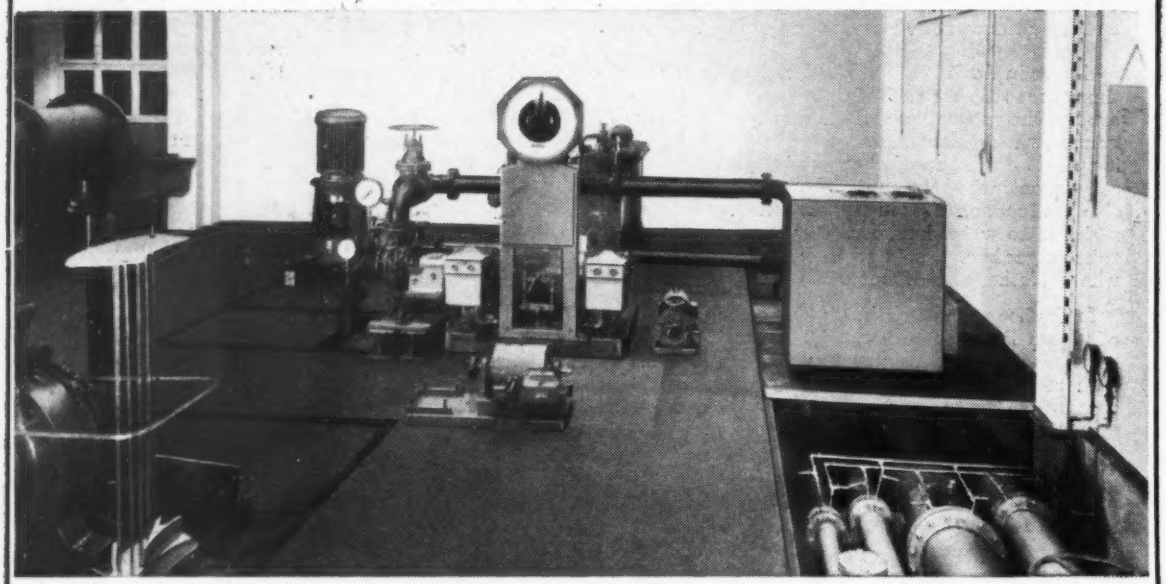

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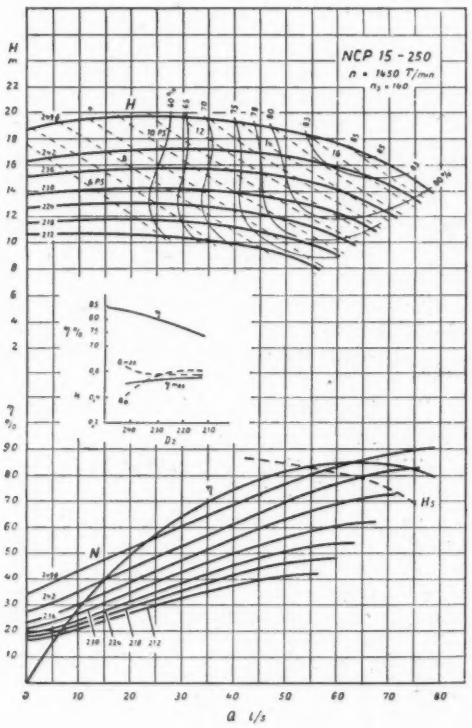

Standardization of Turbomachinery

By VON. ENG. K. RUTSHI

With consideration to future hard competition, it is necessary to rationalize manufacturing of all articles as much as possible and thereby the standardization of the machines is of great importance for the increase of the performance. The standardization in question is very economical and natural when the so called standard coefficients are used.

In the following we will in the first part give you an explanation of the standard coefficients especially with consideration to the trials of Mr. O. Kienzle, and in the second part the practical use of the standard coefficients on standardized centrifugal pumps will be shown.

Standard coefficients

As it is necessary to graduate technical articles of different sizes purposely, the need of these standard coefficients has appeared, and already in the 1870 the Frenchman Renardby his standardization of ropes made the relation between serial numbers equal; in five steps he would arrive at a value of ten times the first. Thus he found the intervals being $5\sqrt{10} = 1.5849$, i. e. each next coming factor in the series is so much bigger than the preceding one. For practical use he found the rounded factors, 1, 1.6, 2.5, 4, 6.3, 10; and for a more accurate graduation he found the values 1, 1.25, 1.6, 2, 2.5., 3.2, 4, 5, 6.3, 8, 10.

Later on other scientists found the same standard coefficients independent of each other, on account of their logical construction. Almost at the same time proposals made in France in Dec. 1921, and in Germany in Jan, 1922 for a standard table, and this resulted finally in standard table DIN 323, corresponding to the American standard table ASA Z. 17.1 or VSM standard table 10050. The table nr. 1 is taken out of the last publication of the standard table DIN 323 and shows in four lines different graduations, within the range of $10n-10n+1$, in 40, 20, 10 or 5 intervals.

$$\begin{aligned}
40\text{th series } x &= 40\sqrt{10} = 10.0593 = \text{about } 1.06\\
20\text{th series } x &= 20\sqrt{10} = 1.1220 = \text{about } 1.12\\
10\text{th series } x &= 10\sqrt{10} = 1.2589 = \text{about } 1.26\\
5\text{th series } x &= 5\sqrt{10} = 1.5849 = \text{about } 1.58
\end{aligned}$$

The standard coefficients given by this table form a system of numerals with quite definite qualities, which can be used as a new technical one. The division of the decade in 40, 20, 10 or 5 members leads to simple only 3-figured *mantisses*, easy to add, (column 4 table 1), and thus the logarithmical table for all standard coefficients is given.

The following calculation rules from the geometrical graduation:

1. Products and quotients of arbitrarily chosen standard coefficients are standard coefficients again.
2. Wholenumbered powers of standard coefficients are standard coefficients again.
3. The double and the half value of each standard coefficient is a standard coefficient again.
4. Each member is the geometrical mean of two members, placed equally as much before as behind in the series, for instance:
   3.552=2.8. 4.5. (20th series).
5. Finally it is very important that the numbers π, 2π, $\pi/4$, $\pi/2$, $\sqrt{2}$, $3\sqrt{2}$ 1 inch and so on, are near to the standard coefficients.

When accustomed to the standard coefficients, it is very easy to make mental multiplications and divisions quickly only by addition and substraction of the mantisses, and the exactitude obtained by the standard coefficients is almost always sufficient for technical calculations, especially as many of the quantities in question only are know with a rather big unexactitude (solidity-factors, empirical values) However none of the values of the standard coefficients differs from its exact value with more than +1.26% resp.—1.01% according to column 6 in the table 1. Which coefficient you may ever choose, the result of the calculation will show, that there is none differing with morethan 3% from the next-coming standard coefficient in the series 40.

It is very easy to fix the standard coefficients with the slide rule (fig. 1) provided that the slide rule has a section A of the logarithmical section B. By setting the runner mark on one of the 10th values of the section A (100, 200, 300 and so on), the corresponding standard coefficients of the series 10 will be obtained on the logarithmical section B. It is very easy to round off the found exact values to head-values. By setting on the half intermediate values of the section A (050, 100, 150, 200...) the standard coefficients of the series 20 is obtained.

It is also possible to make series of standard coefficients on half-logarithmical paper, according to fig. 2. From this figure it will be seen, that beside the fundamental series R5, 10, 20 and 40, also so called intermittent series can be obtained, by omitting single members as for instance

the series	40/3	20/3	10/3
with one interval	X=1.18	1.4	2

The practical use of these half-logarithmical papers consists in the fact that it is possible to introduce either indicated series not yet corresponding to the standard coefficients in order to examine if the present values can be standardized without much change or to examine new series in this way.

On fig. 2, series b, the following values of the inside diameter of the discharge branches in a pump series are indicated: 25, 32, 40, 50, 60 (63), 80, 100, 125, 150, (160), 200 mm. i. d. and thereby their connecting line is parallel to the series R10. Even line a, corresponding to the later mentioned specific number of revolutions n_s by centrifugal pumps, agrees with the series R 20/3 and the line C, corresponding to impeller diameter agrees with the series R 10. It is also possible to use millimeter paper for this logarithmical account, but then the scale of the standard coefficients must be entered.

The need and the importance of this standardization cannot be better explained than by the examples given by Mr. O. Kienzle of standardized and not standardized inside diameter series. If for example different products, i. e. turned parts of steel, independent of each other, are standardized diameter series x, y, z, can appear, according to fig. 3, which in all require 17 different bar diameters with corresponding tools (dies, drills for the corresponding parts, reamers, mandrels, and so on) and templates. A standardization according to the series 20 reduces the numbers of bar diameters to only 11, and hereby the former dimensions of each diameter series x, y, z, can be transferred into the new standard series. In this way the number of tools and other outfit can be essentially reduced.

Besides the diameters, it is also possible to graduate different details according to the standard coefficients, and in our VSM-standards of hand-wheels, vane root handles, pressure intervals in discharge pipes and so on, we will find an extensive development of the system with standard coefficients. However it is much easier to see the advantage of the system by standardising not only parts of machinery but the whole machines their outputs and main dimensions according to the maxim of the standard coefficients. If then one factory is beginning to standardize in this way, other firms producing the same kind of machinery, are following by and by the example and then the same standard graduations are used for the whole group of industry. It is then possible for the employer of the machines to compare the different prices with regard to outputs and sizes. Machines with the same outputs can thus be brought together into groups, even if they are produced by many different makers.

This uniformed shape and graduation according to standard coefficients makes it at the same time possible for the manufacturer to rationalize his types and it is also a way to mutual understanding. There is already a firm basis for the standardization, and when national standards are calculated on these standard coefficients, there are great possibilities for their being used unchanged as international standards.

These standard coefficients have further the advantage of being neutral, and thus, it is now much easier to come to an understanding than earlier, when different values had to be discussed and accepted

The use of Standard Coefficients in the Manufacture of Centrifugal Pumps

(*a*) *Standardisation of output*—

It is comparatively simple to standardize machine tools, such as stamping machines, presses, lathes, etc. of different sizes. The standardization is a bit more difficult when it comes to power engines of unlimited sizes and outputs. So for example centrifugal pumps are built for different capacities, and for a each capacity the head can vary, the number of revolutions being constant or variable. Therefore it is of interest to find a solution that would correspond to the customers demands and to the manufacturers as well.

Of all centrifugal pumps probably the so called single stage low-pressure pumps with volute housings are by far the most common, for which reason pumps of this type should be standardized in the first place. The aim is to cover a great capacity range with a small number of types, though of course, each pump should have a high efficiency within its capacity range.

At first it was necessary to find that basis for the dividing of the total capacity range that would give most suitable gradaution with regard to capacity and head for each size of pump.

It is also essential with regard to manufacturing that as many main dimensions as possible, such as impeller diameters, bore of housings, inside diameter of branches (flange dimensions) should be the same for many types and sizes.

It was essential to apply standard coefficients when determining the impeller diameters. Equal impeller diameters give at equal speed approximately the same heads. Further some different types should always have about the same capacity at different heads, and consequently they should have the same inside diameters on the inlet and discharge branches. If in addition to this various types of pumps could have the same specific revolutions per minute, i.e. geometrically equal dimensions, it would be possible to increase or decrease the size of pumps within a certain series, without much calculation.

Although the solution of the problem as shown in fig. 4, appears to be very simple, it was [1] attained, that fulfilled all requirements. It was, however, advantageous not to apply the actual test results to this system. On the contrary it was with regard to simplicity and comparability, preliminarily better to use one certain value of head [2] H in the approx. correct equation

[1] Necessary to make many compromises before finally the result was.

[2] For each impeller diameter. For this purpose is in fig. 4 for the.

Standardization of Turbomachinery
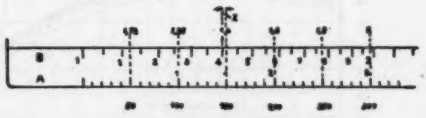
Fig.1.

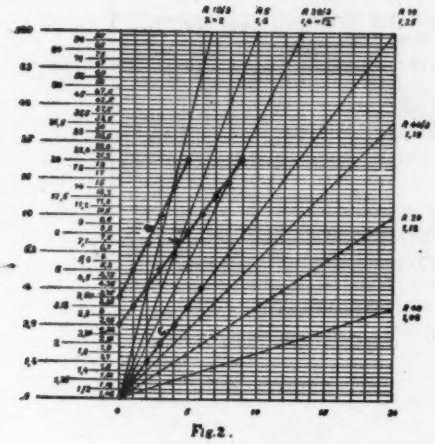

Fig.2.

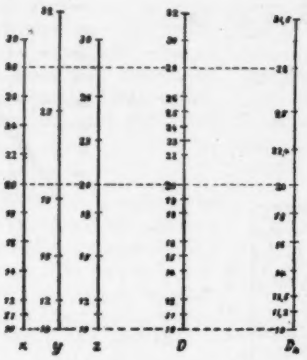

Fig.3.

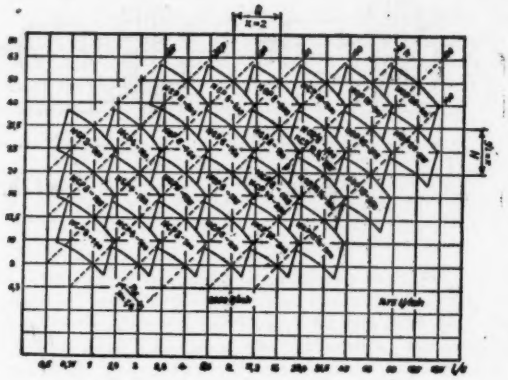

Fig.4.

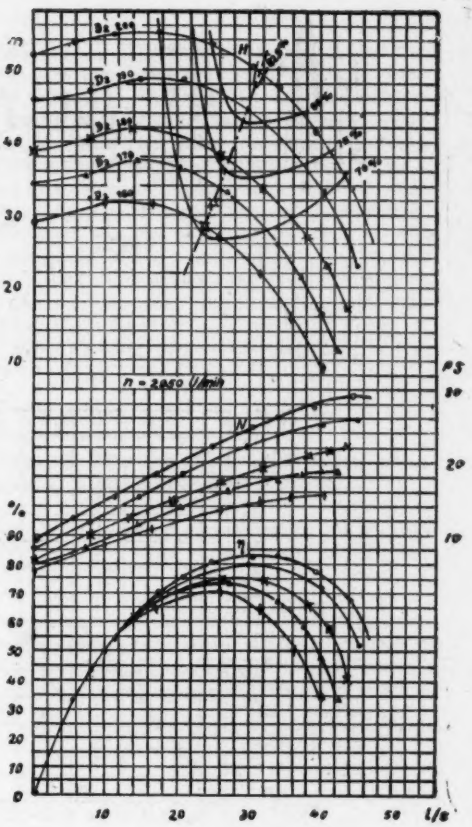

Fig.5.

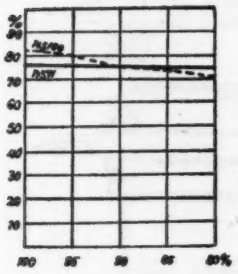

Fig.6.

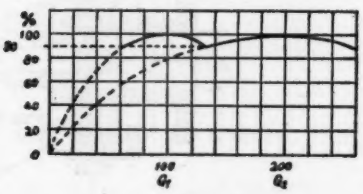

Fig.7.

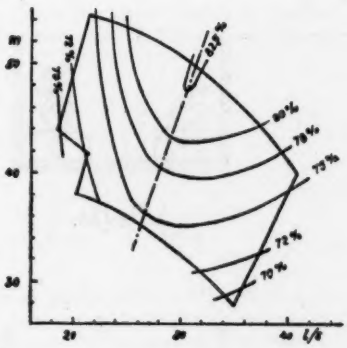

Fig.8.

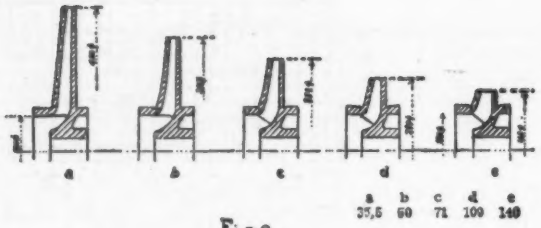

Fig.9.

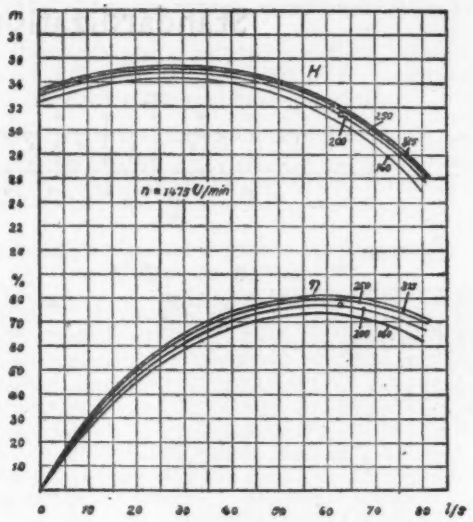

Fig.10.

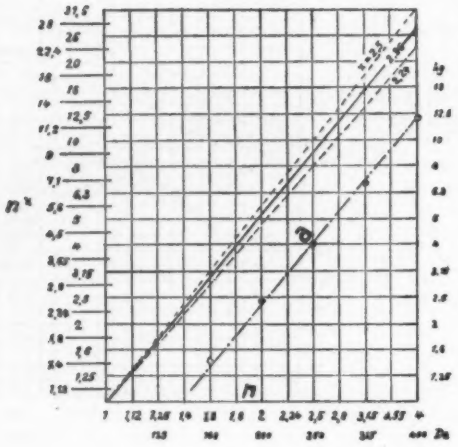

Fig 11.

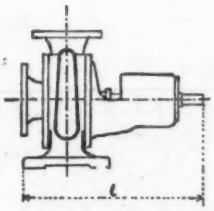

Fig.12a.

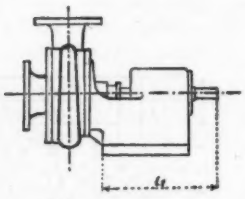

Fig.12b.

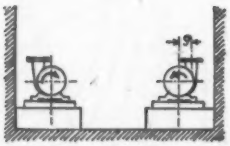

Fig.13a.

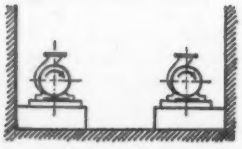

Fig.13b

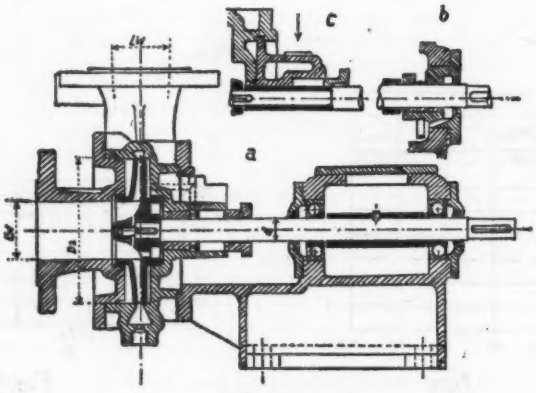

Fig.14.

$$H = \frac{u2^2}{2g}$$

the so called head value v always given the approx. value 1, whereas the measured values of v for the pumps in question vary between approx. 0.95—1.5.

For the determination of u^2 the speeds 2950 rpm resp. 1475 rpm were selected, the values of these speeds compensating the difference between the value 2g and the approx. value 20, and in this case the heads correspond to standard coefficients. It should be noted that the curves plotted in fig. 4 do not represent performance tests, the real data usually being somewhat higher or lower, depending on pump size and specific speed.

For the standardization of head it was necessary to consider of the impeller diameter should be selected from the series R 20, or if the series R 10 was suitable, or if the series between this two had to be used. For this purpose it was necessary to investigate to what extent an impeller diameter could be reduced without influencing too much on the efficiency, so that a reasonable efficiency could be attained throughout the head range of that certain pump. In fig. 5 are plotted the test results for a pump having an original impeller diameter of 200 mm, which diameter was reduced in steps of 10 mm until re next standard diameter was reached. As will be seen from the diagram the capacity and head are reduced more than indicated by the simple formula

$$\frac{D}{D_v} = \frac{Q}{Q_v} \; ; \; \frac{D^2}{D_v} = \frac{H}{H_v}$$

Thus when reducing the head to say 80% of the normal, i.e. down to the next standard head, the impeller diameter should not be reduced 80% but slightly less. Anyhow the difference between the value P^1 and the calculated value P depends on a pondered design of impeller and housing. In general impellers with high specific speed usually show a greater decrease than narrow impellers. In fig.6 is plotted the corrolation between the efficiencies and the reduction if impeller Φ for two impellers with specific speeds n_s =100 resp. 71 (see also fig. 9). By both diagrams is shown that even by reduction of impeller diameters down to the limit range, what often happens in practical life, the efficiencies are rather good. Thus a definite graduation of impeller diameter according to the series of standard coefficient R 10

D_2=100, 125, 160, 200, 250, 315, 400 mm etc.

was chosen. Hereby the diameter of each impeller is about 1.25 times bigger than the former one. It is possible to continue the series quite optionally. However it can be noted that by bigger pumps it is convenient to chose the series R 20 with consideration to the savings in materials especially for the casings.

By graduating the capacity it was evident that the step to the next type did not cause any important decrease of efficiency in the both extreme ranges. In fig. 7, the efficiency of the pump is taken from the diagram in fig. 5, and there the maximum pressure and the corresponding capacity Q_v are indicated with 100, without dimensions. By different trials of comparison it was shown, that for the next capacity Q_2 the solution x=2 was the most suitable one, as well for single pumps as for the whole capacity range, and the efficiency for this capacity range will not be less than 90% of the maximum value. If for example, the smallest type of pump with a capacity of 1 l/sec. is designed, the following capacities according to the standard series R 10/3 are attained:

Q=1, 2, 4, 8, 16, 31. 5, 63, 125 l/sec. etc.

and the same time for a vertical series of pumps as fig. 4 with equal inside diameter of branches.

If, for instance, in the performance field of one sample pump according to fig. 8, the curves co-ordinating the points with equal efficiencies are plotted, it will be shown, that the biggest part of this field belongs to an efficiency range which only differs about 10% from the maximum value. The result attained makes it obvious that the graduation of the capacities as well as of the heads are well indicated, especially when considered that a rather extensive performance field can be covered with a relatively small number of pumps.

If for the special pump the specific speed

$$n_s = 3.65n \; Q^{1/2} \cdot H^{3/4}$$

is calculated, another strangeness of the standardized capacity is obvious, namely; one group of pumps always gets the same specific speed. Thus it is possible to manufacture these groups of pumps according to the design, i. e. to make them larger or smaller, as wanted, and thereby the following equations are valid:

$$H = H_v \left(\frac{n}{n_v}\right)^2 \times \left(\frac{D}{D_v}\right)^2$$

$$Q = Q_v \left(\frac{n}{n_v}\right) \times \left(\frac{D}{D_v}\right)^3$$

At constant speed the head is raised to the second power and the capacity to the third power of the diameter. Hereby the advantages of the standard coefficients are further developed and multiplied with each other they are standard coefficients again. In table 2 the increase of Q and H is shown, which then influences the increase of D_2.

The following types belong to a group of pumps of the same kind with the specific speed $n_s = 100$

NCP 5/100, NCP 6/125, NCP 8/160, NCP 10/200 etc.

and therein the first figure indicates the sequence number of the internal diameter of branch (cm) and the other the sequence number of the outside diameter of impeller, resp. housing bore (mm). Out of the relation between these two values in table 3, it is even possible to see form of the impeller a to e, and thus also the construction or specific speed.

It is also possible to increase the field of application, fig. 4 by enlarging the diameter series, with maintained construction a to e. It is only necessary to use a standardized diameter. By production on a small as well as on a large scale it is advantageous to use the standardized diameters and the specific speeds, and it is thus possible to predict the new design with great accuracy, standard pumps already existing. Comparative trials with these pumps of similar design but of different sizes, form a basis, fig. 10, which makes it possible to predict, with great certainty, the efficiency and performance extrapolation for any degree of enlargement. Of course it is necessary to consider the risks for cavitation by such an enlargement, and to choose a resp. lower speed.

The weight of a big unit can also be calculated easier and more accurately with this method, than it is possible with a detailed methods of calculation, generally used. This is of greatest advantage for projecting and quotation work. If by a geometrical increase, all dimensions should be linearly enlarged the weights would increase in the third power. With regard to resp. thicker walls by smaller pumps, which are necessary from foundry technical reasons, practical experience has shown that the power factor is not 3, but by centrifugal pumps varying between 2.25 and 2.5. Now it is possible to plot in fig. 11. the weights of wellknown sizes for different construction parts (impellers, housings), or for the whole pumps, but even if desired for other machines, and then, it is easy to read off the power factor necessary for the calculation of the weights, with great exactitude. So for example, the weights of impellers are indicated by diameter D_2 and thereby their connecting line a runs parallel to the power factor 2.36. Now it is possible to read off the intermed ate values directly in kilograms from line a or from its lengthening, or it is also possible on basis of the weight of a proto-type, to calculate the weight of a n-times larger unit through multiplication with the corresponding factor n^x.

(b) Standardisation of the design

The purpose of all standardization is, besides a rationalized production, to attain also a uniformity of the types of machine parts and assembled machines, and this makes it important that different manufacturers work according to standard principles. In the case of centrifugal pumps, the single stage volute type pumps are the easiest to standardize, and such pumps are to-day being produced by a great number of manufacturers, all over the world, practically thoughout of a type, shown in fig. 12(*b*). The essential features of this design is that the pump casing is mounted on a flange of a pedestal bearing block, whereas with the older design, shown in fig. 12a the feet are placed on the pump casing, and the bearing bracket flanged on the housing. The main advantage of the new design is that the pump branches can be placed in any position, simply through revolving the pump casing round the shaft centre. This, of course, was not possible on the older type, where the pump branch, pump casing and the feet were parts of the same casting, and thus their position in relation to each other were fixed· For each position of branch relatively to the feet, a new pattern had to be made.

In the new design b, the standardized bearing block can be used for many types of pumps, requiring approximately the same horsepower. The standardized bearing block can, consequently, be produced in greater series, which makes the production cheaper, and for each particular type of assembled pump, the stock of bearing blocks can be kept fairly small.

Another advantage of the new design is that the same bedplates can be used for, say, 5-6 different pump types, having the same size of bearing block, with the standardized length 11. A standardization with regard to the position of the discharge branch is also possible. On older pumps these were usually arranged as shown in fig. 13(*a*), whereas now-a-days a branch arrangement according to fig. 13(*b*) is coming into use. It could be expected that the sharp bend of the discharge volute should give a lower pumpefficiency, and for this reason tests were carried out with pumps having housings of the two different types, but similar impellers. These tests, which were carried out for various specific speeds, showed that practically the same efficiency was attained in all cases. Thus it would be possible to use housing of type b exclusively.

It is often wanted that pump groups are symmetrically arranged and it is necessary in this case, when using pumps of type a, to have two different pumps with different directions of rotation. When using the new type of pump casing symmetry can be easily arranged with pumps running in the same direction, and in addition to this, it is an advantage that the discharge branch of each pump is right over the shaft centre. In pumps of the earlier design the measure generally was not the same for different makes. In pumps built according to the new principles with the discharge branch right over the shaft centre, the single stage volute pumps must necessarily get the same dimensions, especially if different manufacturers use the standard series of performance data, when laying out their pump series.

Fig. 14 shows the pumps with all different parts standardized. The standardized bearing block is provided with two ball bearings with consideration to the arrangement of the impeller, and thereby a central running is warranted even if the sleeve is worn-out. It is also possible to provide the designed pump with two plain bearings instead of ball bearings by

using the same shaft. The whole bearing pedestal, with snaft, sleeve and stuffiing box, forms a standardized unit, which can be used for many different pumps requiring the same horsepower. This standardized seating is graduated for a shaft diameter of 20, 25, 30, 40, 50mm (series 10 x =1.25), and these five designs are completely sufficient for the totally 30 pumps shown on the performance field of fig.4. The housing with the discharge branch radially placed according to fig. 13b is so designed that it is easy to exchange part of the stuffing box for a cooling jacket for temperatures ranging above 115°C. Further it is common practice to provide the shaft of a hot-waterpump with an exchangeable sleeve of stainless steel. Also in this case the standardization makes it possible to use only a few details for this purpose.

For every part of the pump, especially the housing, a low weight is aimed at, of course, without deteriorating the structural strength. This low weight is especially important in pumps made of stainless steel and bronze but even for the cost of transportation the low weight is of interest.

For rationalized production the permissible deviation (tolerance) for important dimensions is reduced to a minimum, but in order to faciliate manufacturing a deviation of ± 0.25 mm is permitted for any other dimensions. If earlier, when turning the external djameter of the impeller, the turner has tried to attain the measure 159 mm, the deviation statement of to-day tells him, that for this impeller diameter the measure indicated can be passed with ± 0.25 mm, and thus he saves a lit of time. This deveiation is permitted, especially as the clearance between the external diameter of impeller and the bore of the casing is sufficient, and this small inexactitude of the impeller diameter cannot influence the output.

As for the bores of casing for the appliance of the suction end cover, only small deviations are permitted, the measure D_2=100, 125, 160, 200, 250 mm being selected and the impeller diameter being at the same time 1 mm smaller only. As for different pumps always the same impeller diameters and casing bores appear, and also measure-and drill-templates are very often used. It is also possible to use the bolt hold diameter of pipe flanges for the cosing bores in order to keep the number of templates as small as possibe.

Conclusions

In addition to the above fields of application of single stage volute pumps it is also possible to use the standardized performances and designs for the production of centrifugal pumps for any other kind of design. If the series a in fig. 2 is further developed the following standardized speeds n_s =31.5, 50, 71, 100, 1+0, 200, 280, 400, 560, 800, 1120, for the whole field of application is obtained.

They correspond to the following shapes of impeller:

31.5 to 140 = radial impeller with singlecurved vane

200 and 280 = radialimpeller with doublecurved vane

400 and 560 = half-axial pump impeller

800 and 1120 = axial pump impeller (propeller)

By the standardization it is possible to use only above eleven specific speeds instead of an infinite number of different shapes. Each of the mentioned standardized speeds is 1.4 times bigger than the preceding one. By these standardized speeds the number of the corresponding shapes of impellers is essentially reduced, and a very careful development of the design is enabled. It is then possible to increase or decrease the design for each wanted size.

The small deviations of the test results from the range of performance put down in fig. 4 on the basis of simplified presumptions, which are due to the degree of accuracy in machining etc., ought to be taken into the bargain. This is only fair, taking into consideration the fact that the uniformity of manufacture of machines made by different works result of this standardization is rewarded by better machining and as finish, a somewhat higher efficiency, or a slight increase in output. Generally speaking, you may expect, at specific speeds of up to n_s =100, especially inathe case of big pumps, a slight increase of the curve velues, and at specific speeds above 100 a steadily decrease of the same values.

It is, of course, obvious that in the case of big single plants the pumps could be designed for exactly the pumping data required, instead of for standardized data, but also in those special cases it would be advisable to base the design as much as possible on standard coefficients, especially with regard to the impeller diameters, using, perhpas, a series of coefficients with smaller intervals.

On about the same basis not only all centrifugal type machines, such as turbines, fans, and compressors, but also any other type of machines could be standardized, taking into consideration, of course, the special purpose for which they are intended.

Together with a rational manufacture, the creation of unitary stages of output and size will always be desirable, and the standard coefficients are, indeed the best means by which this aim can be gained.

Cavitation in Hydraulic Turbines and Pumps

By K. SEETHARAMIAH

Every designer of prime mover for the generation of power is striving hard to design and manufacture bigger and bigger units, with higher and higher speeds in order to make the units more and more economical in its first cost and in its maintenance. In trying to achieve this the designer is faced with varied problems. A steam turbine designer is faced with metallurgical limitations in regard to rotor and blading materials, the gas turbine designer with the problem of materials which can withstand higher and higher temperatures. For a hydraulic engineer, for a given head and flow of water there are very few limitations and the most troublesome problem is that of Cavitation. This problem has further been increased due to the trend in the design of very high speed turbines and pumps working under adverse conditions. The cavitation problem is not only faced in turbines and pumps but is being faced in marine propellers, feed pumps, submarines, underwater ballistics, etc. In this paper it is proposed to describe the cavitation phenomena, its effects and the various methods by which the effect of cavitation could be reduced if not completely eliminated.

Historical Background on the Cavitation Phenomena

Cavitation is by no means a mysterious phenomena and is a very natural and explicable one. Very little was known about this before, as most of the machines that were used then were of low speeds and small capacities. However, Euler in 1754, while developing hydro-dynamic equations for turbines had indicated that we have to guard against cavitation. Later the eating of the propeller blades of trans-Atlanic liners gave impetus for the study of this phenomena towards the end of the last century and considerable investigations on this phenomena was taken up at the turn of the present century. Considerable work is being done to understand the phenomena of cavitation, effects of cavitation and their prevention. Research on this problem is being extensively done in U. S. A., U. K., Europe, Japan. A start has been made at the Indian Institute of Science, Bangalore, for basic studies on this phenomena of cavitation and its effects. Different types of equipment like the accelerated cavitation set up, venturi type of set ups for cavitation inception and cavitation damage and a water tunnel, have been built in the laboratory here for this study.

Cavitation Phenomena Defined

Cavitation is caused by the fundamental property of water to evaporate at a sufficiently low pressure even when the temperature is quite normal. This results in the formation of bubbles in the liquid which may contain vapour or gases in it. These bubbles expand and when the pressure outside attains a fairly large value, they collapse and the collapse of these bubbles result in cavitation pitting which will be explained in detail later. Cavitation is also defined as the formation of voids within a body of moving liquid (or around a body moving in liquid) when the particles of liquid fail to adhere to the boundaries of the passageway. The failure of the particles to adhere to the boundaries occurs when there is insufficient internal pressure to overcome the inertia of the particles and to take sufficiently curved paths along the boundary. When these bubbles come into a region of higher pressures they collapse and if they collapse on a solid boundary, pitting will be the result. Thus the cavitation phenomena is a dynamic one and is caused in any flow of water wherein the pressure is dropped to very low values due to a sudden or gradual increase in velocity, consequent to constriction in water passages.

Physical aspects of Cavitation

Cavitation is a dynamic phenomenon, controlled by hydro-dynamics of flow and the principle properties of the liquid, which commences once the pressure is reduced to vapour pressure. Any increase in the temperature depended upon vapour pressure, or any localised pressure reduction may give rise to a cavity form. In the case of turbines cavitation is mainly due to flow around a bend and/or sometimes vortex flow where the local pressure reduction occurs. This is further accelerated due to the sharp curvatures and local roughness and sepration and pressure drop from the leading edge to the trailing edge. The various theories on cavitation and cavitation damage will be dealt with later.

Types of Cavitation

There are different types of cavities met with in practice and they are: (1) Transient cavities or bubble cavitation where small individual bubbles grow, sometimes oscillate and eventually collapse and disappear, (2) Steady-state cavities. This is also known as sheet cavitation and laminar cavitation. These are the stationary cavities observed behind blunt bodies, at the sharp leading edges of hydrofoil profile. (3) Non-stationary cavities. These are similar to steady-state cavity but vary, with time. (4) Travelling cavities. These cavities go through several cycles of rebound and collapse where as in the fixed type of cavity large sections of it are entrained by the flowing stream of liquid. As opposed to type of cavitation, we have the kinds of cavitation.

Prof. Shalnov of the U. S. S. R. has made another classification which is more applicable in hydraulic machines. The classification is based on the location

of different points in a machine. Taking Kaplan turbine as an example, he calls the cavitation on blades as "profile or face cavitation". "Blade clearance cavitation" occurs between the blade face and the inner surface of the guide vanes or the draft tube. "Blade tip cavitation" occurs at the tip of the blades. "Roughness cavitation" occurs due to the perterbances due to the bad design and due to bad workmanship of the blade surface. The blade shoulder cavitation takes place at the junction of the blade and the hub.

The stage of cavitation or the phase of its growth is characterised by the extension of the cavitation zone or by the value of the cavitation criterion. Any kind or type of cavitation may be encountered at various phases of its growth depending upon the flow velocities and absolute Pr. Value.

Different Theories on Cavitation Inception

Various investigators have given various explanations for the inception of cavitation. It is universally accepted that cavitation is the formation, growth and collapse of cavities in water due to pressures far below the vapour pressure at a given temperature. Some investigatiors have attributed this to chemical effect, some to electrolytic affect, some take it as thermodynamic phenomena. But the latest accepted theory is the nucleus theory which postulates that some nucleus which may be in a solid, liquid or gaseous phase, is essential for the formation of cavities.

Tensile Strength of Liquids

In an absolutely pure water, the only way of creating cavitation is by producing tension in it. To fracture a liquid creating a gap between molecular layers of the order of one molecular radius, the work required to be done is of the order of 10,000 atmospheres. Experiments have shown that cavitation can occur at very much reduced pressures. This has been attributed due to the presence of nuclei within the liquid containing air or vapour or both. It has been experimentally proved that water saturated with air but denucleated by application of high pressures exhibited high tensile strength. This is a further proof that the presence of nuclei is necessary for the inception of cavitation at pressures of the order of vapor pressure. The presence of a solid or a third phase is indicated by the fact that the surface energy of a bubble bounded by a solid surface and a liquid surface may be very low as a consequence of which cavitation may be expected to begin above as well as below the vapor pressure, depending upon the size and number of the nuclei.

It has also been seen that the presence of large amounts of entrained air and of surface active agents may raise the cavitation pressure well above the vapour pressure. But latest experiments have shown that it is size, distribution and quality of undissolved gas nuclei that govern the pressure at the inception of cavitation rather than the total air or gas content in the fluid.

Surface active materials reduce surface tension at surfaces in interior of liquids or at boundaries. They encourage early formation of cavities. The surface active materials are absorbed on the surface of the materials and this increases the interfacial tension. Thus the inception of cavitation depend on the nature of these surface active materials.

Shear flow in several phases of cavitation has been a subject of study. In laminar boundary layer, the inception of cavitation is on the body itself. The bubbles emerge and grow at the point of origin itself and reach a size and then are removed and swept downstream by the flow. In case of turbulent boundary layers, where the pressure gradient is not very great the inception takes place in a region of 5 to 21% or the boundary layer thickness and in region of adverse pressure gradients inception occurs at about 30% of the boundary layer thickness.

In the wakes of bodies cavitation depend both on velocity and pressure fluctuations that occur in such turbulent flow.

Effects of Cavitation

The effects of cavitation in hydraulic machines are:

1. Fall in the efficiency of the machines.
2. The vibrations.
3. Cavitation noise.
4. Cavitation damage.

The fall in the efficiency of hydraulic turbines due to cavitation is a very serious factor and this is due to the cavitation damage of the material of the runner. The bursting of the cavitation bubbles will result in undesirable noise and this noise level will have to be kept at a minimum, as otherwise it will have very bad effects on the operating personnel. The collapse of the cavitation bubbles being instantaneous result in vibrations, the intensity of vibrations may be so great that the vibrations may result in bad cracks in the foundations and the penstocks. All the above effects of cavitation are to be minimised if not completely eliminated.

The Mechanics of Cavitation Damage

Damage in Hydraulic machines may be due to corrosion, erosion or cavitation pitting. The corrosion is due to chemical action or electrolytic action of water on the material of the machine. Erosion is due to the removal of the material due to the impacts of suspended solid particles on the metallic surface and consequent pealing of the material. The cavitation damage otherwise known as cavitation pitting is defined as the actual erosion of material under the pressures produced by the collapse of the bubbles formed by cavitation. It is possible to have cavitation without pitting but there can be no pitting without cavitation.

In order to protect the components that are likely to get the cavitation attack, economic considerations dictate operation close to or tolerable conditions of cavitation and sometimes the repairs are accepted as necessary evil. But the turbines and pumps have

to operate outside the design range resulting in pitting of runner casings and liners.

There are several theories propounded by several investigations on the mechanics of cavitation damage. Cavitation damage has been conceded today as due to mechanical action Lord Raleigh has calculated the pressure for the collapse of an empty cavity within a liquid at rest and they are very high and the failure is due to fatigue. The thermodynamic theory also considers the high pressures developed and consequent fatigue failure. Jet impact theory takes into consideration that water droplets are accelerated by the collapse of energy to very high velocities and impinge on the boundary surface. Haller has done some work in this direction and has shown that repeated drop impacts of moderate magnitude can destroy the material.

Effects of corrosion in cavitation damage is still under investigation. Possibility of corrosion due to crystalline deformation is also considered.

Prof. R.T. Knapp has related the damage to individual bubble collapse of small cavities, travelling along the cavity interface to implode at the rear of the cavity where the zones of maximum damage are found. A cyclic process of cavity collapse and reforming oscillalates the stagnation point over an area. This concept links the damage to stagnation pressure. Prof. Knapp has also proved that the total damage may prove independent of the length of the cavitation zone whereas damage intensity may be found greatest for short cavities. It is evident that the damage intensity is controlled by the pressure available to collapse the cavity and the pressure controls the cavitation zone.

Shalner has shown that the erosion intensity is linked to Reynolds Number based on the diameter of the cylindrical piece. He also observed that zones of maximum damage are claimed to occur at the point of bubble formation and crack away from the surface of the test piece.

It is now an established fact that the pitting of materials under cavitation is essentially a fatigue process under continuous impact of blows which ultimately result in the chipping of the surface. On this roughned surface the action is accelerated due to stress concentrations. Thus it has to be realised that the neccessity for proper selection of materials for the turbo-machines and their components cannot be underestimated.

The Cavitation Indices

In hydraulic machines the consequences of cavitation, viz., loss of material, damage, loss of performance, noise and vibrations are the most undesirable. It is not economical to design completely cavitation free and to allow a certain amount of cavitation damage. Model studies are made for cavitation conditions as a function of operating conditions and applied to prototypes. In all stages cavitation is pressure controlled, as such, to clarify cavitation, pressure is a very important factor.

For steady flow conditions assuming that the loss between any two points in a stream flow is negligible and the effect of gravity is nil. We can write down the Bernoulli equation

$$P_\infty + \frac{\rho V_m^2}{2} = P_m + \frac{\rho V_m^2}{2}$$

where ∞ and m denote the points referred to in fluid and P and V the pressure and velocities at the points.

We can also write down

$$2.\frac{P_\infty - P_m}{PV_\infty^2} = \left[\frac{A_\infty}{A_m}\right]^2 - 1 = K.$$

Where A and A_m are the areas at points where the pressures and velocities are measured.

Introducing the term vapour pressure P_v we can write:

$$2.\frac{P_\infty - P_v}{\rho V_\infty^2} - 2\ \frac{P_m - P_v}{\rho V_m^2} = K.$$

The term $2\ \frac{P_\infty - P_v}{\rho V_\infty^2} = \sigma$ is called the Cavitation parameter. Cavitation will commence when $P_m = P_v$

Hence $-\sigma_{cri} = 2\ \frac{P_m - P_v}{\rho V^2 \propto}$

$\sigma = 2\ \frac{P_\infty - P_v}{\rho V_\infty^2}$ is the cavitation parameter for Venturi test section.

In the case of hydrofoil sections

$$\sigma = 2\ \frac{P_\infty - P_v}{\rho V_\infty^2}$$

Some investigations use the symbols σ_{min}, σ_{nor} and σ_{cri}. σ_{min} will enable to find out the minimum pressure at which cavitation could be avoided. σ_{nor} This gives an indication of the pressure reserves against cavitation and σ_{cri} indicate when the pressures below vapour pressures are reached.

For turbines and pumps we can write down:

$$\sigma - \sigma_{cri} = \frac{P_{min} - P_v}{H_{T.p.}} = \frac{M}{H_{T.p.}}$$

$P_{min} - P_v$ represent the pressure reserve against occurrence of critical cavitation at the point of lowest pressure in the machine.

M = Margin of safety in ft.

$H_{T.p.}$ = Head utilised by turbine or Head developed by pump. Sigma critical can also be expressed as

$\sigma_{cri} = \frac{H_a - H_{s\,max}}{H_{T.p.}}$ This is the Thoma definition of Cavitation parameter.

Where H_a=Barametric height

H_s = Critical static suction height.

$$\sigma \text{ cri} = \frac{H_a - H_{s\,max} - \frac{P_v}{\gamma}}{H_{T.p.}}$$

$\frac{P_v}{\gamma}$ Vapour pressure of liquid.

If $H_{s\,max}$= Max permissible static runner elevation above the lowest tailrace or sump level.

We can write down the safe setting of the machines as

$H_{s\,max} = Ha - \triangle H_{.T.p.} - M$ for a machine with horizontal shaft.

and $H_{s\,max} = Ha - \triangle H_{.T.p.} - \frac{D_e}{2} - M$ for a horizontal shaft.

$\triangle H_T$ and $\triangle H_p$ is the minimum pressure head required at the runner eye to avoid critical state of cavitation at the point of lowest pressure in runner.

$$\triangle H_T = \frac{P_v}{r\gamma} + \lambda_1 \frac{C_2^2}{2g} + \lambda_2 \frac{W_2^2}{2g}$$

and $$\triangle H_p = \frac{P_v}{\gamma} + \lambda_1 \frac{C_1^2}{2g} + \lambda_1 \frac{W_1^2}{2g}$$

Pfleiderer has derived another equation for cavitation parameter

$$S = \frac{Q}{K \triangle H^{3/2}} \left(\frac{N^2}{100} \right)$$

Q= flow in Cm/sec.

$\triangle H = \sigma H_{.T.p.}$ in Metres.

N = R. P. M.

$K = 1 - \frac{Dh}{De}$ where D_h and D_e are the hub and eye diameters.

S is a function of angle of attack and of λ factors. S depend on N_s the specific speed and head $H_{.T.p.}$ in the range of low and medium specific speeds.

$S = \frac{N\sqrt{Q}}{\triangle H^{3/4}}$ and for the use of centrifugal pumps.

$$S = \frac{N_s}{\sigma^{3/4}}$$

The term S is called the suction specific speed and this is used in comparing the suction performance in turbo machines of different N_s by relating Cavitation performance to runner suction and suction or discharge branch shape.

Another parameter used as cavitation parameter is called the thermal cavitation parameter and this is not much in use.

$$B = \frac{V_v}{VB} \quad \frac{\triangle hf}{L} = \frac{V_v}{V_L}$$

Where V_v and V_L are specific volumes of vapour and liquid.

$\triangle hf$ = enthalpy per unit weight of liquid (expressed in thermal units)

L = Latent heat of vaporisation.

V_v and V_L are respective vapour and liquid volumes.

The general formula extensively used in cavitation analysis is:

$$K = \frac{H_L - H_V}{V^2/2g}$$

K = cavitation parameter

H_L = absolute pressure in liquid in feet of liquid.

H_v = vapor pressure of liquid corresponding to the temperature in ft. of liquid.

V = relative velocity between the body and the liquid in ft. per sec.

g = acceleration due to gravity in ft./sec/sec.

The sigma value for the hydraulic turbine is:

$$\sigma = \frac{H_b - H_s - H_v}{H}$$

H_b = Barametric pressure in ft. of water

H_s = Static suction head on bottom of the turbine runner in feet of water

H = Net operating head on turbine in ft.

Mr. K. K. Shalnev of the Institute of Mechanics, Academy of Science, U. S. S. R. has developed the following equations for clearnance Cavitation.

Clearance Cavitation Coefficient.

$$H_a = \frac{P_\infty - P_v}{\gamma q}$$

Where P is the pressure upstream of the clearance and beyond its zone of influence

$q = \frac{V^2}{2g}$ where V is the mean vellocity in the clearance

γ= specific weight of water

P_v= Vapour pressure of water.

Similarly Shalnev has developed equations for surface roughness Cavitation, tip cavitation, profile cavitation.

In the case of pumps the head required to prevent cavitation is:

$$H_{sr} = \frac{V_e^2}{2g} + A\left(\frac{U_1^2}{2g} + \frac{V_e^2}{2g}\right)$$

Where H_{sr} =total energy required at the centreline of the impeller in excess of the vapour pressure to prevent cavitation at the pump impeller inlet region expressed in ft. of liquid.

V_e=liquid velocity just at the inlet but not yet within, the impeller, ft. per sec.

U_1=Peripheral velocity at the impeller inlet computed from the same diameter used to compute impeller eye area and hence V_e.

A=Diamensionless no., a factor varying with the angle of attack.

g=acceleration due to gravity.

Prevention of Cavitation in Hydraulic Machines ("Consequences" has already been mentioned in the earlier pages):—

The result of cavitation in hydraulic turbines and pumps are:

1. Loss of performance
2. Material Damage.
3. Vibration
4. Noise.

The efficiency of the machine as the cavitation sets in drops down considerably and there will be the drooping characteristic. The drop in efficiency in low specific speed machines is attributed to the reduction in the mean runner passage pressure being reduced to vapor pressure. In the case of axial flow machine with high specific speeds, the effects are far more gradual since runner passages are replaced by lifting surfaces, moving through a free stream, with cavitation the lift decreases and drag increases. In the case of low specific speed axial flow machines the water passages get blocked which in tern modify virtually the blade and affect the performance of the machine.

In designing for optimum cavitation conditions it is essential to avoid sudden contour changes in guide surfaces upstream of the runner and in the casing since separation or eddy formation are conducive to cavitation. The lowest pressures occur in the runner and thus the design of the runner is a very important factor.

In the case of pumps the cavitation is due to the abnormalities in the local surface, causing departures from the ideal shapes, and the entrance conditions causing a non-anticipated flow pattern and non-uniform flow in the impeller by condition on the discharge side of the runner. The pressure drop at the leading edge of the vane depends upon the velocity of approach of the vane in the fluid, angle of attack and conversion of the pressure head to velocity head at the inlet of the impeller.

Material Damage

Both in pumps and turbines cavitation damage has been extensively met with inspite of best design conditions. Cavitation is observed on guide vanes. speed rings, tounges of spiral casings, in runners and in the throat of draft tube, splitters of draft tube. The possible damages are due to contour separation, roughness, clearance flow. The roughness cavitation is localised in individual obstacles. Clearance flow in axial flow machines can cause severe pittings at the blade tip and on the suction face of the tip region and also in runner casing.

Vibration

The vibration of turbines and pumps is a very undesirable factor and this, if transmitted to penstock and pipelines, reaches the resonant frequency, it is very dangerous. This should be avoided. The vibration may be due to blade cavitation or runner hub cavitation. Draft tube surges are considered due to hub vortex cavitation. This type of vibration can be reduced by properly designing trailing edge. Injection of air has been found to be very effective. This acts as a cushioning and this method could be adopted only in case of emergency as admission of air may lower the efficiency.

Noise

Another objectionable effect of cavitation is the cavitation noise. This is related to highly developed stages of cavitation In power plants and pumping stations this will be a nuisance for the working staff and should be suppressed if not completely eliminated. Admission of air is only an emergency measure.

Choice of Materials for Turbo Machine Components under Cavitation Attack

Field experience has shown that cavitation pitting may occur in machine operating under normal load without much drop in efficiency. The damage under these conditions may be due to surface roughness, contour shape, clearance flow and could be noticed as due to clearance flows and separation of flow. In a model test this cannot be reproduced such that the absolute safeguard against cavitation cannot be done.

The cavitation damage can be reduced by

1. Adequate surface finish
2. Correct contour design
3. Cavitation resistant material.

In the case of impulse turbines the parts that are affected by cavitation are the nozzle, the spear.

Buckets are damaged to a certain extent by cavitation but largely by erosion due to presence of sand, ice and other solids. The splitters of the bucket are likely to be damaged and these are the portions to be protected.

In the case of Francis turbines the parts which are susceptible to cavitation damage are the guide vanes. Wearing rings, suction face of runner blades, draft tube throat wheel rims and wearing rings.

In the case of Kaplan turbines the parts primarily affected are the runner blades, the runner hub and draft tube and runner casing.

In the case of axial flow pumps the cavitation damage is likely to occur on blades, hub and casing and in centrifugal pump, the parts affected by cavitation are the runner blades, the eye of the runner, shroud and the casing.

If proper care is not taken to make proper design and suitable material for the runner a lot of time and money will have to be wasted for overhauls and the period of shutdown will be long. But by using proper cavitation resisting materials the interval between overhauls can be increased.

A lot of investigations have been conducted all over the world to find out the best material which gives minimum cavitation damage. The process of cavitation damage is quite slow. Sometimes, different simple materials are welded to the runners and tested. Sometimes model runners are made of different materials and tested. Sometimes jet impact tests are made. The quickest and best method is the use of magnetostriction oscillator for the study of the cavitation damage of materials.

Experiments conducted have shown that practically all materials are affected by cavitation but the intensity of damage varies.

Cast iron as cast is more susceptible to cavitation damage than stainless steel with different proportions of chronium and nickle. Experiments conducted on accelerated cavitation had shown loss in material expressed as mm^3 at 20°C for cast iron 636.00 as against 1.3 for 17/7 stainless chronium-nickle steel.

The most desirable properties required for the materials used for machines are: (1) Adequate mechnical properties, (2) Resistance to erosion attack, corrosion fatigue, cavitation attack. The materials should also have good casting and machining property and easy repair conditions in the field.

It has been an accepted fact that the cavitation damage is mainly a mechanical action and is entirely due to work-hardening of metal surfaces and the type of failure observed is entirely due to fatigue. Any material used in hydraulic machines should possess the following physical and mechanical properties to withstand the effect of cavitation. The material should have adequate stiffness, should be capable of taking very fine polish, should be fine grained, high fatigue strength, good corrosion resistance and should be hard.

Cast iron is no good to resist cavitation damage and it cannot be welded. If welded after pre-heating; the material becomes brittle. This material could be used for low head machines in scroll tube, guide vanes, small capacity pumps and draft tubes.

Low carbon steel could be used upto 500 meters head but this material has poor cavitation resistance and used for medium head units.

Stainless steel with 12 and 14% chronium has very good cavitation resistance and also stand against erosion attack, can be welded after preheating. This material is extensively used for Pelton, Francis and Kaplan runners.

Of the several materials and alloys stainless steel has proved the best material in all its properties. It stands well against attack from foreign matter can stand chemical action, can be work-hardened. Aluminium, Bronze alloys like Ampco Bronze has very high cavitation resistance and is extensively used. Bronze runners are good but casting is difficult and welding troublesome.

Economic considerations and the difficulty in thermal treatment of stainless steel has led to the use of cast low carbon and low alloy steel, with prewelds.

Compared with cost of low carbon steel, the stainless steel is nearly 2½ times costly and carbon steel with prewelding of stainless steel is about 1½ time costly.

With regard to welding methods, the two well known are the deposit welding and the strip welding, and strip welding is better as there is no possibility of deformation or residual stress.

It has become an accepted practice to fabricate the machines out of stainless steel than casting as it is less expensive.

Use of paints, stainless steel sprays rubber is also used and the stainless steel spray has proved effective. Use of stellite and welded stellite overlays are quite good but expensive.

Liquid Neoprene is used with very good effect for field repairs and it has been found to be economical and even superior to chromium steel. The binding techniques has made the use of neoprene very promising.

In the foregoing paragraphs the various materials used and the treatment to reduce cavitation damage has been explained in detail. The cavitation effects could considerably be reduced by a regular inspection, welding of the pitted portions, resurfacing of the blades and vanes. A sub-layer of 25/20 or 15/12 stainless steel followed by a second layer of stainless steel 18/8 or 19/9 has been found to be effective.

Another method of reducing the cavitation damage is by keeping the machine as low as possible. In the equation of Thoma No. $\sigma = \dfrac{H_B - H_S - V_P}{\rho V^2}$ it could be seen that if H_s the height of setting is small, the Thoma number will be big and this means the cavitation susceptibility could be considerably decreased by keeping the sigma value high which can be done by decreasing H_s but there is a limit for this as the cost of keeping the runner very low will be quite considerable and may not be economical.

The Flow in the Rotating Passages of a Centrifugal Pump Impeller

By G. A. J. YOUNG

When endeavouring to understand the mechanism of the transfer of energy from the impeller to the fluid in a centrifugal pump, it is useful to have a knowledge of the velocity distributions within the impeller passages. When these velocity distributions are known, deductions can be made about the pressure distributions along the blade faces and hence the blade loading and input work. This indicates those portions of the blades where improvement of shape could best be made and the location and extent of cavitation danger.

There are two basic methods of observing the flow within the passages of a rotating impeller. Pitot tubes or other instruments may be rotated with the impeller and the indication of speed and direction taken out through the shaft. Alternatively, the paths of tracers of unit specific gravity introduced into the flow in a transparent model impeller may be recorded photographically. This latter method has been used by other investigators to obtain records of instantaneous absolute flow paths and has now been extended by us to obtain photographic records of both the relative and absolute instantaneous velocity vectors.

The method and apparatus have already been described in B. H. R. A. Publication No. RR. 518, (1) but for convenience diagrams and photographs of the apparatus are reproduced here as Figures 1 to 4, and a brief description of the technique is given below.

This paper is an extention of the work described in RR. 518 and includes records of the flow at half normal, normal and one and half times normal delivery. Normal delivery is defined as the delivery corresponding to the best efficiency for the complete pump; i. e. a flow ratio, C_{m2}/U_2 of 0.091 and a head coefficient, gh/U_2^2 of 0.53. (Fig. 2).

THE TECHNIQUE

The impeller draws the water from a vertical suction pipe and discharges it between parallel plates into a large cylindrical tank which is connected by 6" pipework to a booster pump, throttle valve and metering nozzle back to the suction pipe (Fig. 1). Solid particles of unit specific gravity are introduced into the suction pipe well upstream of the impeller. Records of the relative flow paths of these tracers are obtained by exposing a photographic plate in a camera attached to and rotating with the impeller shaft, for say 1/50th second at an impeller speed of 60 r. p. m., and arranging for two high intensity flashes to occur one just after the opening and the other just before the closing of the camera shutter.

Records of the absolute flow paths are obtained by a similar technique using a stationary camera.

Sample records of the flow paths are shown in Figs. 5 to 9.

For convenience in the photographic work the tests have been made at an impeller speed of 60 r. p. m. At this speed the impeller Reynolds Number (as defined by Stepanoff [2]) is 7,200 (cusecs/ft. dia. $\times 7$). Stepanoff has shown that head coefficient is unaffected by speed above an impeller Reynolds Number of 3,000, so that the bulk of the flow pattern measured at these low speeds should still be representative of high speed conditions. Overall efficiency however does increase with increasing Reynolds Number. Other B. H. R. A. tests verified that relative velocity was proportional to impeller speed down to the same critical impeller Reynolds Number of 3000.

THE IMPELLER

The impeller is a full size model, constructed in "Perspex", of a 14.13 inch (360 mm) diameter impeller used in a commercial 6" centrifugal pump having a normal duty of 970 g. p. m. and 132 ft. head at 1450 r. p. m. The specific speed is 1158. Dimensions of the impeller and the H-Q characteristic of the complete pump are shown in Fig. 2.

PRESENTATION OF RESULTS

(a) Observed Relative Velocities:

The distributions of relative velocity throughout the impeller passages at 0.5 normal, normal and $1.5\times$ normal delivery are shown in Figs. 10 to 12. These are composite diagrams obtained by superimposing the records from a number of plates, and are useful in obtaining a general picture of the flow patterns at the three rates of flow. There appear however to be variations between the records obtained from different plates taken at the same nominal flow rate, so that subsequent analyses have been made from individual plates at each flow rate.

The distribution of relative velocity across a blade passage (B-C in Fig. 16) at the three flow rates are shown in Figs. 13 to 18. These curves have been extrapolated to the suction face of the blade in order to estimate the loss—free relative velocities at the blade faces which are necessary if pressures are to be calculated from the relative Bernoulli equation. In cases where there is a region of low velocity near the trailing face of the blade, such as sections 6, 7 and 58 of Figure 15, the graph has been extrapolated as a straight line

and the higher velocity used in the computation of pressure distribution.

The distributions of relative velocities along the blade faces derived from the extrapolations of the above graphs are shown in Figs. 19 to 21.

(b) Derived Quantities

The pressure distributions along the blade faces have been calculated from the relative velocity distribution by using the Bernoulli equationfor rotating co-ordinates.

$$P/\gamma + \frac{W^2}{2g} - \frac{U^2}{2g} = H_w = \text{Constant} \qquad (1)$$

Assuming the relative total head, H_w, is constant and equal to the total head at inlet P_T/γ, the real total head, H_e, may be shown to be equal to the relative total head, plus the head generated by the impeller.

$$H_w = \frac{p}{\gamma} + \frac{W^2 - U^2}{2g} = \frac{P_T}{\gamma} \text{ the total head at inlet.}$$

$$H_e = \frac{p}{\gamma} + \frac{C^2}{2g}$$

$$= \frac{p}{\gamma} + \frac{W^2 \sin^2 \rho}{2g} + \frac{(u - w\cos\rho)^2}{2g}$$

$$= \frac{p}{\gamma} + \frac{w^2 \sin^2\rho + u^2 + w^2\cos^2\rho - 2uw\cos\rho}{2g}$$

$$= \frac{p}{\gamma} + \frac{u^2 + w^2}{2g} - \frac{2uw\cos\rho}{2g}$$

$$= \frac{p}{\gamma} + \frac{u^2 + w^2}{2g} - \frac{2u(u - C_u)}{2g}$$

$$= \frac{p}{\gamma} + \frac{w^2 - u^2}{2g} + \frac{u.Cu}{g}$$

i. e. $H_e - H_w = \frac{u.Cu}{g}$, the head generated by the impeller up to the radius r where the peripheral velocity is u.

A non-dimensional coefficient representing the static pressure at a point at radius r in the impeller may be defined as,

$$C_p = \frac{p - P_T}{\gamma u_2^2/2g} = \left[\frac{u}{u_2}\right]^2 - \left[\frac{w}{u_2}\right]^2$$

$$= \left[\frac{r}{r_2}\right]^2 - \left[\frac{w}{u_2}\right]^2 \qquad (2)$$

Using suffixes (D) and (T) for the driving and trailing faces of the blade, respectively, the difference of pressure across the blade at any radius r is given by.

$$\Delta p = \gamma \frac{u_2^2}{2g} - \left[C_{p(D)} - C_{p(T)}\right]$$

$$= \left[\frac{w}{u_2}\right]^2_{(T)} - \left[\frac{w}{u_2}\right]^2_{(D)} \qquad (3)$$

The blade loading, or perpendicular difference of head across the blade at any radius r is

$$\frac{\Delta p}{\gamma} = \frac{u_2^2}{2g}\left(C_{p(D)} - C_{p(T)}\right) \qquad (4)$$

The torque due to the blade loading for blades is given by

$$T = n \int_{r_1}^{r_2} br\,\Delta p\,dr \qquad (5)$$

$$\text{The impeller power} = 2\pi \frac{N}{60} n \int_{r_1}^{r_2} br\,\Delta p\,dr \qquad (6)$$

The following fuctions have been plotted in Figs. 20 and 21, expressed in terms of the peripheral velocity head $u_2^2/2g$

Kinetic energy of the relative flow at the trailing face $= \frac{W^2_{(D)}}{3g}$

Kinetic energy of the relative flow at the trailing face $= \frac{W^2_{(T)}}{2g}$

The static pressure head (referred to the total head at inlet)

at the driving face $= P_{(D)}/\gamma$

The static pressure head (referred to the total head at inlet) at the trailing face $= P_{(T)}/\gamma$

The blade loading $= \Delta P/\gamma$

The torque per unit radius (for one blade) $= b\,r\,\Delta p$

DISCUSSION OF RESULTS

Figures 15 to 18 show that at normal and 1.5 × normal delivery the relative velocity along sections across the passage normal to the flow increases approximately linearly from the driving face of one blade to the trailing face of the preceding blade except, at normal delivery, on the trailing face near the blade exit tip, where there is a region of low relative velocity as shown at sections 6-8 of Fig. 15. This indicates a wake or region of low energy fluid such as has been observed by Mr. Worster [3], but there is no evidence from the photographs to suggest the presence of back eddies on the outlet tips of the trailing face of a blade as has been found by Mr. Peck [4].

The loss of energy due to this bad flow amounts to about 4% of the total energy at section (8) of figure 16 or 25% of the kinetic energy. In terms of the kinetic energy this loss is the same at sections 6, 7 and 8. So that it would appear that all the loss has occured before reaching section (6) and that there is no further loss after this section.

There is no evidence of this wake at the higher flow rate (Fig. 17).

The impeller at present has flat tips and it would be an interesting experiment at a later date to endeavour to relate the increase in head which is normally obtained by backing-off the blade tips on the trailing face with the changes in relative velocity distribution and flow pattern. Mr. Lupton (5) suggests that backing-off the tips causes a local increase in the whirl component. Alternatively it may be that the backing-off of the blade tips reduces the wake on the trailing face and thereby increases the flow component.

The low relative velocity on the driving face of the blade and the higher velocity on the trailing face is in general agreement with the flow patterns deduced from potential flow theory (5). The distributions of relative velocity along the blade faces, obtained by extrapolation of the relative velocity distributions across the passage, are shown in Figures 20 and 21 for normal and one and a half times normal deliveries. These have been used in calculating the pressure distributions along the blade faces from the Bernoulli equation for rotating co-ordinates, as explained in section 4(b). From these the pressure across the blade or blade loading, and input work have been derived.

Within the accuracy of the measurements, which is about 10%, the derived blade loading and input work agree with that obtained from the head-quantity tity characteristic for the complete pump for both normal and 1.5× normal delivery. This is not very conclusive evidence in itself, but it supports the view put forward by others (6), (7) that the Bernoulli equation can be used to relate pressures and relative velocities in a centrifugal pump, when the flow rate is high and there are not extensive regions of separation and consequent losses of energy.

The mean blade loading at normal delivery derived from Fig. 20 is about 0.18 $u_2^2/2g$ and the ratio of the maximum to mean blade loading about 1.5. The maximum blade loading and torque/unit radius occur at 5.3 and 5.6 inches radius respectively which is the region just beyoud the blade overlap.

By plotting the total work done up to a given radius, derived from the blade loading, against radius (Fig. 22) it will be seen that 62% of the total work is done between 4" and 6" radius whereas only 18% is done up to 4" radius, 20% in the last inch of radius and 8% in the last half inch.

At 1.5× normal delivery the mean blade loading is about 0.16 $u_2^2/2g$ and the ratio of the maximum to mean blade loading about 1.6. The maximum blade loading and torque/unit radius occur at about 6 inches radius.

Fig. 22 shows that at this delivery, although the same fraction (62%) of the total work is done between 4" and 6" radius only 12% is done up to 4" radius, 26% in the last inch of radius and 8% in the last half inch.

For comparison with Fig. 22, the distribution of head generated throughout an impeller passage has been plotted in Figure 23. This has been prepared from the product of whirl component of absolute flow C_u and the local peripheral velocity of the impeller u.

The four lines O, P, Q and R refer to the four sections of the passage shown in the sketch.

This diagram indicates that at a given radius more head is generated near the driving face of the blade than near the trailing face, except near the blade outlet tip where the revers occurse. This again emphasizes the influence of the shape of the blade trailing edge at the tip upon the head generated and suggests that the low relative velocity and hence high absolute whirl component in this region assist in increasing the mean head generated at outlet, although the resultant peaks in the absolute flow may cause additional mixing losses in the casing.

The mean head coefficient at the impeller periphery determined from the average of the products of C_{u2} u_2 at the impeller periphery is 0.52 which is a little below the head coefficient for the complete pump. (0.53 at b.e.p.) The impeller head coefficient should be slightly greater than that for the complete pump because of the losses in the volute casing.

At half normal delivery the relative flow patterns are complex, as will be seen from Figs. 9, 13 and 19. The velocities are higher on the driving face of the blade than on the trailing face and there is evidence that the flow differs from passage to passage. The transverse velocity distributions shown in Fig. 13 are obtained from one plate and are given as an example of the non-uniformity. The relative velocity distributions along the blade faces shown in Figure 19 are from three different plates and illustrate the scatter which exists in the results.

Morelli (6) has found similar instability in the flow at small deliveries in tests on a radial impeller and has explained it as stalling of alternate vanes. A similar phenomenon has been observed in investigation of surges in centrifugal compressors (8).

CONCLUSIONS

The results so far obtained indicate that the method can provide a valuable insight on the relative and absolute flow patterns within the impeller passages and the effect of blade profile. At normal delivery and above, when velocity distributions are fairly steady, the pressure and total head generated may be derived from the velocity distributions with reasonable accuracy.

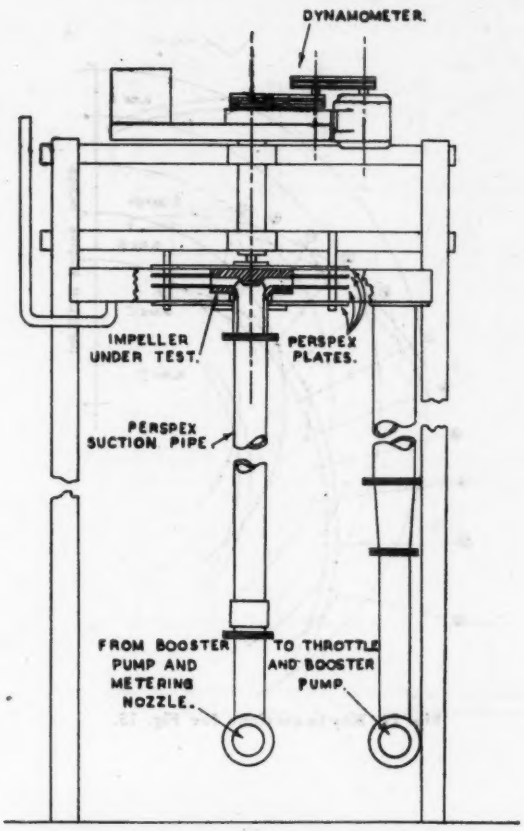

Fig. 1. Diagramatic arrangement of transparent impeller test rig.

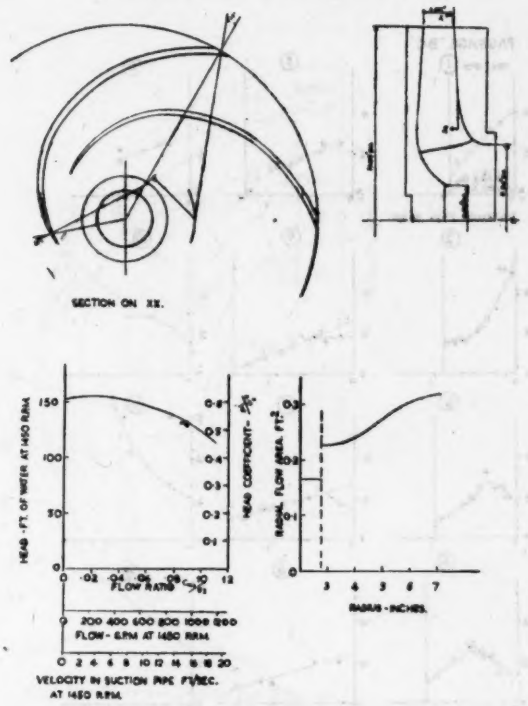
SECTION ON XX.
HEAD - FT. OF WATER AT 1450 R.P.M.
HEAD COEFFICIENT
FLOW RATIO
RADIAL FLOW AREA. FT.²
RADIUS - INCHES.
FLOW - G.P.M. AT 1450 R.P.M.
VELOCITY IN SUCTION PIPE FT/SEC. AT 1450 R.P.M.

Fig. 2. Details of impeller.

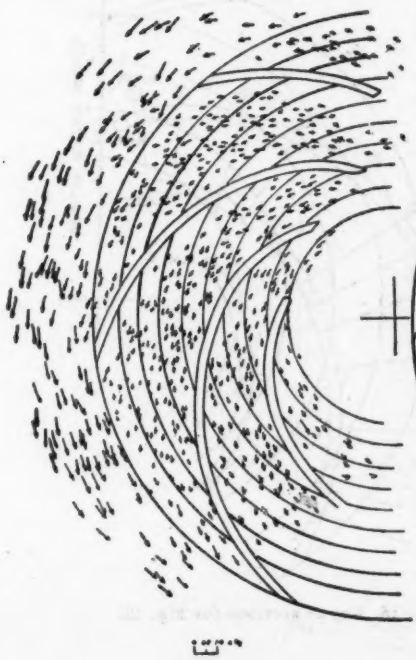

Fig. 10. Relative velocity distribution throughout impeller passages at $\frac{1}{2}$ normal delivery.

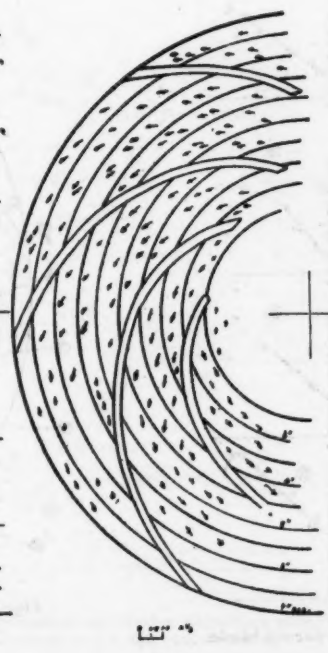

Fig. 11. Relative velocity distribution throughout impeller passages at normal delivery.

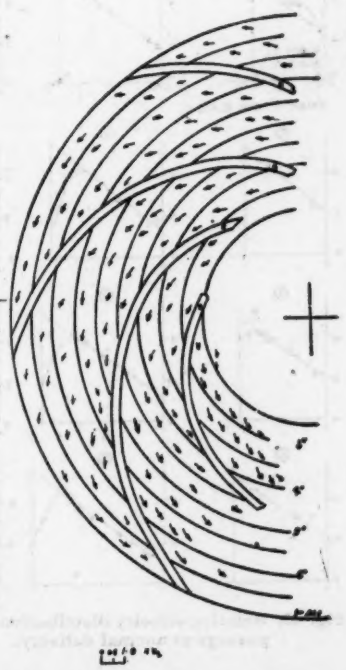

Fig. 12. Relative velocity distribution throughout impeller passages at $1\frac{1}{2}\times$ normal delivery.

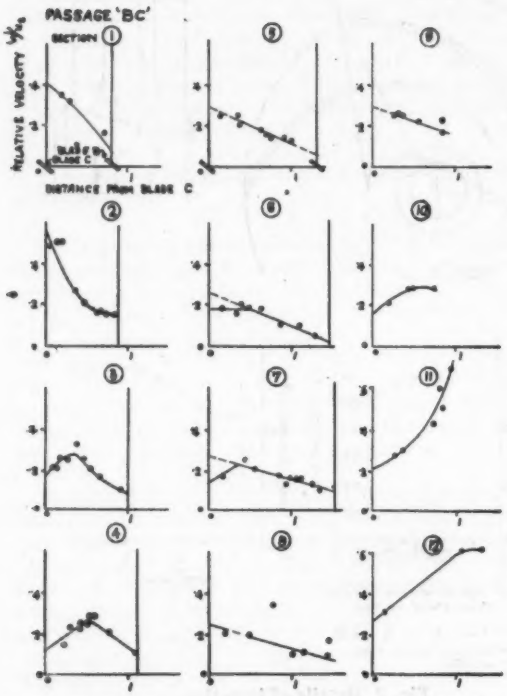

Fig. 13. Relative velocity distribution across blade passage at ½× normal delivery.

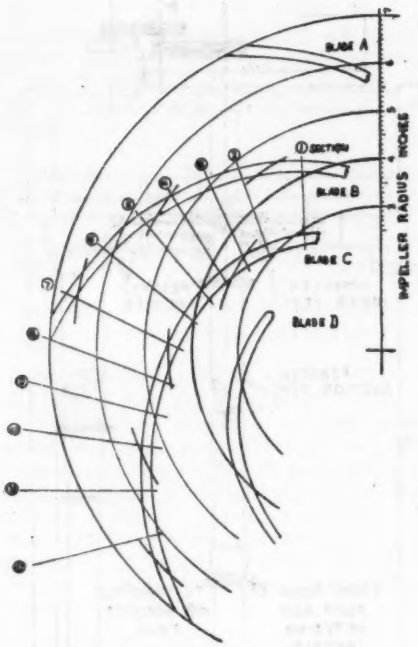

Fig. 14. Key to sections for Fig. 13.

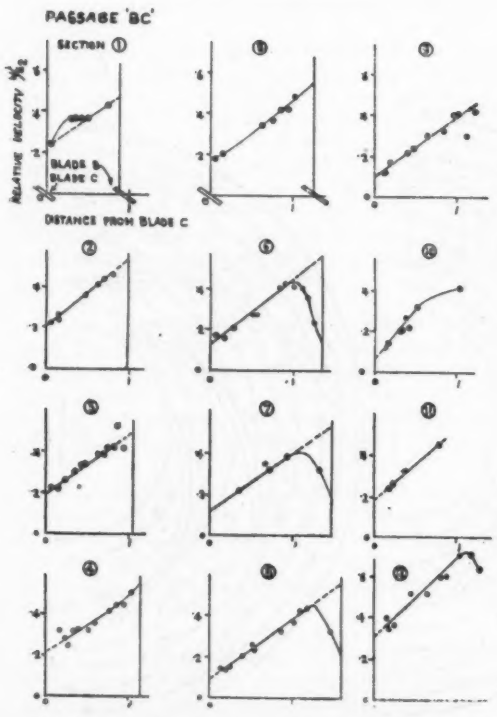

Fig. 15. Relative velocity distribution across blade passage at normal delivery.

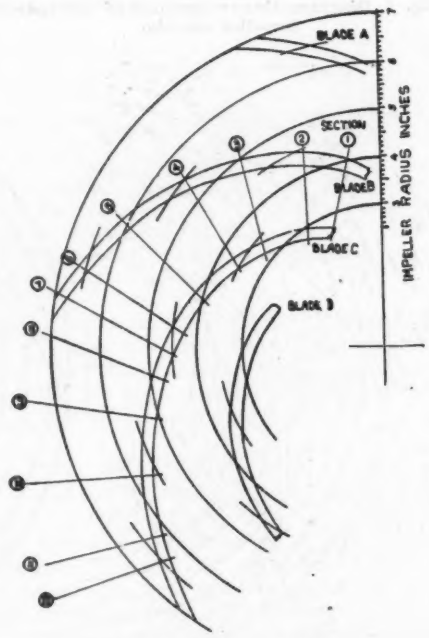

Fig. 16. Key to sections for Fig. 15.

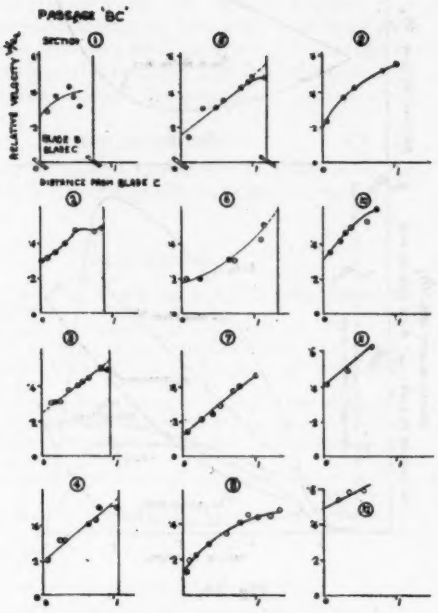

Fig. 17. Relative Velocity Distribution across Blade Passage at 1½× normal delivery.

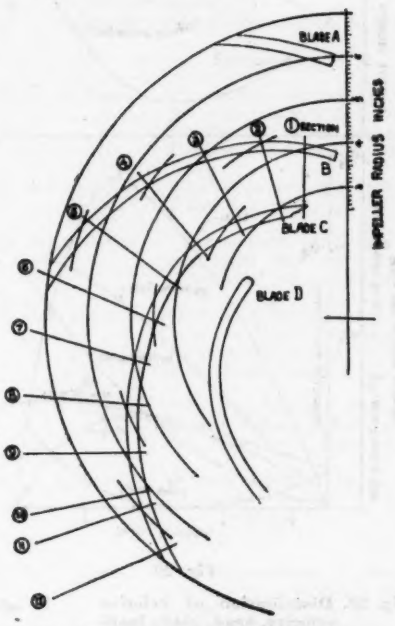

Fig. 18. Key to section for Fig. 17.

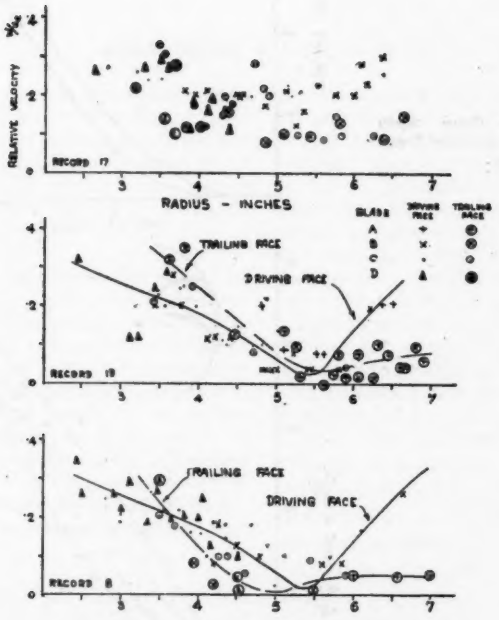

Fig. 19. Distribution of relative velocities along blade faces at ½× normal delivery.

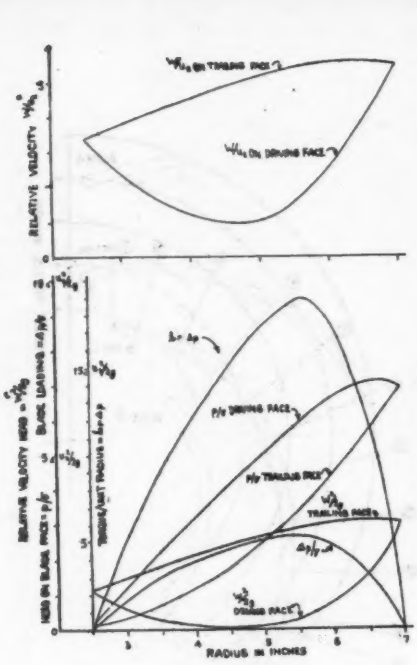

Fig. 20.

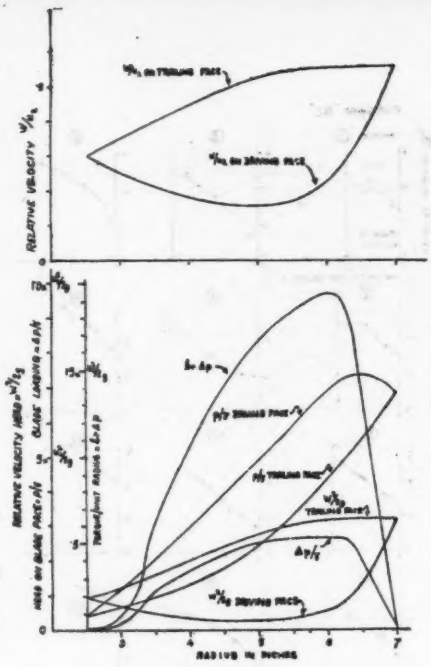

Fig. 21,

Fig. 20. Distribution of relative velocity, head, blade loading and torque along the blade faces at normal delivery.

Fig. 21. Distribution of relative velocity, head, blade loading and torque along the blade faces at 1½× normal delivery.

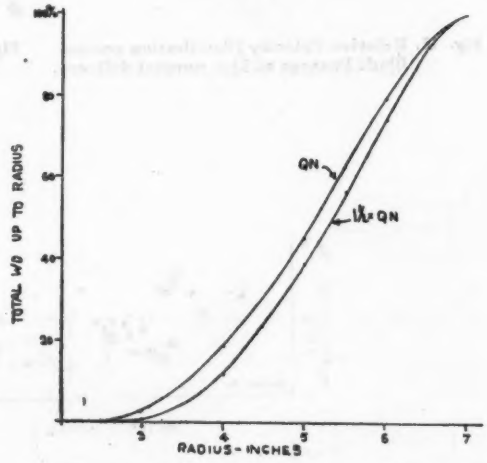

Fig. 22. Total work done upto any radius derived from blade loading.

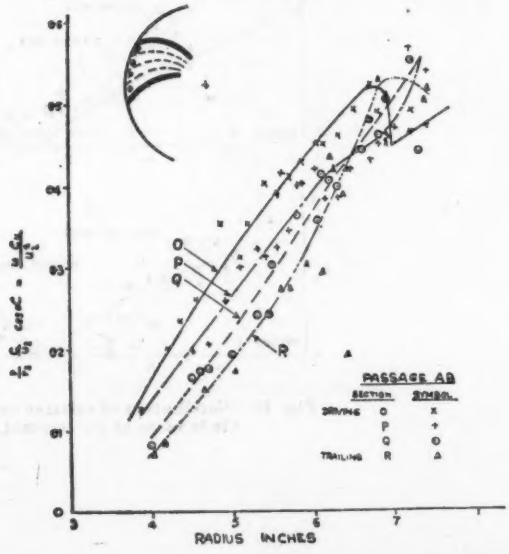

Fig. 23. Distribution of head generated throughout an impeller passage at normal delivery.

At half normal delivery the results confirm the already known instability of flow both with time and between blade passages and suggest that cinematograph recording of the relative flow in all the passages at the same time would be desirable.

Acknowledgements

The Author wishes to thank Mr. L. Fellerman for his assistance in obtaining and analysing the results.

List of Symbols

- b = width of impeller, at any radius r from front shroud to back shroud.
- C = absolute velocity of particle at any radius r.
- C_u = the whirl component of the absolute velocity. The suffixes 1 and 2 refer to the eye and the periphery of the impeller respectively.
- C_p = a non-dimensional coefficient representing the static pressure at a point at any radius r in the impeller, the suffixes (D) and (T) refer to the driving and trailing faces of the blade respectively.
- H_c = the total head in the Bernoulli expression for the absolute flow.
- H_w = the total head in the Bernoulli expression for the relative flow.
- N = speed of impeller, (r.p.m.)
- n = number of blades.
- P = the static pressure at any point in the impeller.
- $\triangle P$ = the perpendicular (normal) difference of pressure across the blade at any radius r.
- P_T = the total pressure at the inlet to the impeller.
- T = Torque due to the blade loading.
- r = the radius of any point in the impeller.
- u = peripheral velocity of impeller at any radius r.
- w = relative velocity of particle at any radius r.
- α = the absolute flow angle.
- ρ = the relative flow angle.
- γ = the specific weight of the fluid (62.4 lb/ft3 for water).

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Continued from page 99]

Speed Control of a Prime Mover

well away from any possible forcing frequencies, such as the firing frequency of a given cylinder. In any case, the magnitude of the swings will be minimized by the introduction of the maximum feasible amount of damping. This is done in an alternator by the use of fully connected amortisseur windings in the field pole faces.

It should be noted that the natural frequency of oscillation of an alternator on synchronous load depends upon its inertia or WK^2 and the synchronizing torque, which in turn depends upon the design of the alternator, the load carried, and the characteristics of the transmission line and paralleled generating equipment. This implies that a range of frequencies must be avoided when designing springs drives or considering possible forcing impulses. This fact makes doubly important the utilization of maximum damping in the amortissur windings so that oscillations which do occur may be of minimum amplitude and die out as quickly as possible.

There is another aspect to this problem in which the governor manufacturer is vitally interested. The manner in which any control system behaves is determined by the dynamic characteristics of the governed system. The purpose of governor adjustment is to so determine the governor characteristics, that, when combined with the other system characteristics, satisfactory control results. This implies first that the system is stable; that is, it returns to the desired equilibrium speed after a disturbance; and second, that the speed (or load) deviation is not too great and that the error is corrected quickly and without excessive oscillation. It follows therefore that a change in system characteristics such as described above should be accompanied by readjustment of the governor for best results. Since this is not usually feasible, the adjustment used is necessarily a compromise. In general a compromise can be found that is acceptable for all normal conditions of operation, but occasionally a unit has such bad paralleled characteristics that satisfactory performance both on and off the line is impossible with a single set of adjustments.

Lubrication of Modern Hydraulic Turbines

By P. GOVINDAKRISHNAYYA

When I was asked to take part in this symposium I felt that the best contribution I could make was to cover a topic based on my long experience in the Petroleum industry. Lubrication of Hydraulic Machines suggested itself but this by itself would be a vast subject and therefore to limit the scope of my paper I shall deal only with the "Lubrication of Hydraulic Turbines".

In the maintenance schedule of plant and machinery 'Lubrication' comes foremost. As is well known maintenance may be of two kinds, preventive and ad-hoc. Lubrication comes under the category of preventive maintenance the benefits of which require no special emphasis.

In this paper I shall try and cover briefly the following aspects:

1. Some fundamentals of friction and lubrication;
2. The various parts of typical hydraulic turbines which require lubrication and their special features;
3. The common methods provided for lubrication of the various parts of the turbines;
4. Lubrication requirements of hydraulic turbines and how these can be adequately looked after.

Fundamentals of Friction and Lubrication

The force that tends to retard the relative motion of two surfaces in contact with one another is called friction. Lubrication is the means by which that force can be reduced or partially overcome.

The aim of lubrication is to prevent metallic contact between the two moving surfaces by interposing a fluid film between the two. Any fluid is a lubricating agent to a lesser or greater degree. Air, water, tallow, fatty oils, mineral oils, soap and graphite are all lubricants. There are literally hundreds of grades and combinations of these lubricants in the market; plus many more created by the addition of special purpose additives, in the regular range of lubricants. The problem is to find the particular lubricant whether oil or grease, best suited to a set of operating conditions. Grease is nothing but soap thickened oil. The function of the soap is to serve as an oil container but it is the oil alone which serves as a lubricant.

Friction is of three kinds; solid, fluid and rolling. These have descending values of co-efficient of friction and therefore in all machinery the aim of the designer is to replace the former by either one of the later two; mostly fluid friction.

Fluid friction itself is of two kinds, 'thick film' or 'hydro-dynamic' or 'thin' film or 'boundary friction. In Hydrodynamic lubrication viscosity is the all important factor as far as the lubricant is concerned. Boundary lubrication is brought about by decreasing the thickness of the oil l film to less than 1/40000" by increasing the load or decreasing the relative speed of the moving parts or the viscosity of the oil; in which case the effect of boundary forces are no longer negligible as compard with the effect of the viscosity of the oil.

Equation $F=2\pi^2 D/C.\ ZN/P$ is a general formulae for the coefficient of friction of a 'Journal bearing'

Where D = Diameter of the journal and C = the diametral clearance between Shaft and Journal

Z = Viscosity, N = Velocity, P = Pressure (load per unit of projected area) F = Coefficient of friction.

The coefficient of friction in 'solid' friction for the usual bearing materials (Dry-nonlubricated surfaces under ordinary conditions) vary between 0.1 to 0.2.

The coefficient of friction in 'Thick Film' or hydrodynamic lubrication normally varies from 0.01 to extremely low values such as 0.001. A fair figure to assume is 0.005 depending upon the viscosity of the oil. It may however go as high as 0.05 depending on the sheology of the lubricant and other operating conditions.

The coefficient of friction under conditions of efficient boundary lubrication may normally be taken at 0.05 to 0.15 but here again the values can vary widely, particularly on the higher side.

In rolling friction, i. e., with ball and roller bearings, the coefficient of friction may be 0.0015 for near optimum conditions and is practically constant through a wide range of speeds and loads.

Some factors to remember in respect of friction are:

1. It is impossible to make a surface which is truly flat or truly round;
2. The true area of contact in any bearing is very small. It varies with the load. Therefore even for comparatively light loads the pressure at the *point of contact* may exceed several hundred tons per square inch;

3. There is evidence that friction of metals is due mainly to the welding of the metal under these enormous pressures at metallic junctions and to the force necessary to shear and break these tiny junctions;

4. The friction between truly clean surfaces are enormous but ordinarily all surfaces are covered with a thin layer (microscopically thin) of moisture and/or oxide. Otherwise movement would have been impossible on this earth;

5. To overcome friction, work has to be done. This energy is dissipated mainly in the form of heat; this also raises the temperature of the surfaces unless cooled to keep the temperature within safe limits;

6. Movement also brings about wear. Under ideal conditions in thick film lubrication there should be negligible wear, since the surfaces do not come in contact;

7. In thin film lubrication (boundary lubrication) the surface layers have a habit of touching. The thin boundary film of oil (a few molecules thick) is unable to cushion, the moving surfaces effectively. This film no longer acts like a liquid; its viscosity is no longer an effective factor. Friction under these conditions is dependent on the nature of the surfaces themselves and on the chemical constitution of the lubricating oil layer;

8. Under boundary lubrication conditions, a mineral oil is not so good a lubricant as a fatty oil of the same viscosity.

There are fundamentally three kinds of motion in machines, i. e., 'Round and round' or 'Rotary', 'To and fro' or 'Reciprocatory' and 'Swinging' or 'Oscillatory'. Care has to be taken in lubrication to allow for these different kinds of movement. It will be found that all moving parts in a machine are supported by machine elements called bearings. Bearings not only support but also restrict the motion of the moving parts to the required degrees of freedom. Since relative motion occurs at such bearings under condition of metallic contact, all bearings have to be lubricated. There are different kinds of bearings but of these the most important for our purposes are the journal bearing, the thrust bearing, and the guide bearing.

For each bearing a suitable method of dispensing the lubricant is incorporated to ensure that the oil is delivered to it as required. Also suitable provision is made in the bearing to ensure that the oil spreads on the rubbing surfaces and builds up sufficient thickness as to effectively separate the moving surfaces.

Bearings are designed for 'thick film' lubrication wherever possible in order to take advantage of the low rate of abrasive wear and the low friction loss but this is not always possible and some bearings have perforce to be designed or have to work under what are known as boundary lubrication conditions or even extreme pressure conditions.

The type of lubricant to be used will depend upon the physical feature of the bearing such as dimensions, design, types of metals used in construction and also its operating conditions such as speed, load, temperature, atmospheric conditions, etc., But generally it will be found that the higher the speed the smaller will be the clearance and the 'lighter' i.e., less viscous the oil for satisfactory operation. On the contrary the slower the speed, the greater will be the clearance and the 'heavier' the oil required.

Viscosity of a lubricant is the most important characteristic in selecting the lubricant for a bearing. For each application there is a particular viscosity which gives optimum conditions of service. Viscosity changes with temperature, with pressure and with ageing and deterioration in use. An optimum viscosity which allows for the effect of all these factors would best suit the bearing even though it might be a compromise between conflicting benefits.

Next come, characteristics like chemical stability, oiliness, detergency, viscosity index, pour point, demulsibility and a host of others. The reason why petroleum derived lubricants are universally for lubricating machinery and plant is because of their inherent stability, easy availability and economy in use, as compared with alternative lubricants like fatty oils, fats, etc., which were effectively displaced by mineral oils more than a century ago. Fatty oils on the other hand possess greater 'oiliness' than mineral oils and for this very reason where necessary blends of the two known as 'compounded oils' are used. Other desirable characteristics which require emphasis according to the application are reinforced by adding suitable substances, generally organic chemicals to minerals oils. Such chemical are called additives and the blends are called 'Additive Oils'. Some of the common characteristics which are reinforced in this manner are detergency, oiliness, oxidation stability, viscosity index, pour point, and soon.

Mineral oils possess inherently many of the desirable characteristics to a lesser or greater degree, depending upon their origin and the refining processes to which they were subjected. Additives are added only where the operating conditions demand them.

It is therefore obvious that for the same application dozens of grades of different qualities are available in the market. It is important that the user and the supplier must ensure that the oil possessing the correct combination of characteristics is chosen for any given application. The choice of a lubricant is more than a mere 'Hit and Miss' proposition. Basically there are three important steps in the selection of a lubricant.

1. Study operating conditions;
2. Determine lubricant characteristics exactly suited to those operating conditions;
3. Select a lubricant;

Most established oil firms have technical departments to provide this service to the user but he, the user, must place due emphasis on price and quality

consistent with economy before making a final choice. The use of low priced lubricants may necessitate frequent oil changes and even renewal of parts due to abnormal wear and tear. Maximum reliability of plant operation is also a factor that is worth taking into account.

The Various Parts of Typical Hydraulic Turbines Which Require Lubrication and their Special Features

Modern hydraulic tubines comprise—

(*a*) the Impluse type and

(*b*) the reaction type. The best representative of the impulse type is the Pelton Wheel. There are however two distinct kinds of turbines covered by the reaction type in modern practice—normally the Francis turbine and the Propeller turbines of which the Kaplan is the most sophisticated and best known. Propeller turbines are of two kinds; one in which the runner vanes are fixed and the other in which the runner vanes are adjustable and the Kaplan typifies the latter variety.

Normally Pelton Wheels are horizontal but for very large cizes it is more convenient and economical to make them vertical. The same is true of Francis turbines. On the other hand Kaplan turbines are generally vertical.

The large majority of hydraulic turbines are installed in hydel stations and invariably the prime mover is directly coupled to the generator and its exciter. The design and installation of all these units are normally carried out as one integral unit. Therefore as far as lubrication is concerned the parts to be lubricated on hydraulic turbines are not distinct from those of the turbo-generator set as a whole.

There are some hydraulic turbines employed for driving industrial machinery direct, but these are comparatively small in size and comparatively few in number. In any case the lubrication features of these industrial water tubines do not differ materially from those installed at hydel stations.

The number of parts to be lubricated in a hydraulic turbine are comparatively few and the operating conditions are not particularly arduous. In turn, lubrication requirements of these machines are therefore not particularly complex. The parts to be lubricated comprise sessentially of main bearings which hold the shaft of the turbines, some subsidiary bearings and a number of miscellaneous parts on the operating and control mechanism of the turbine. Very often the governor, the servomotor and automatic control gear forms part of the lubrication system at a hydel station but since there is a separate paper on 'hydraulic turbine governors' the mechanical details of this equipment, or its oil requirements will not be gone into in this paper.

Common Methods Provided for Lubrication of the Various Parts of the Turbines

In what follows a classification will be made of the various types of bearings on hydraulic turbines and the methods provided for its lubrication.

Horizontal Turbo Sets:—The rotor is mounted horizontally on pedestal bearings with a thrust bearing where end thrust has to be taken care of—as in the case of single runner Francis turbines. The lubrication system may be any one of the following:—

i. For oil lubrication: 1. Circulating system;
2. Ring or collar oilers;
3. Drip-feed oilers.

ii. Grease lubrication: Grease cup for small units.

Vertical Turbo Sets:—The main shaft in this case is suspended from a *Thrust* bearing of the collar or tilting pad type such as the Michell or Kingsbury bearing. To fix the vertical alignment of the shaft there are generally three or more guide bearings; one each immediately above and below the generator and one on the turbine shaft close to the turbine runner or revolving unit. The thrust bearing carries the entire weight of the runner and also the hydraulic thrust on it. The function of the guide bearing is only to hold it in position. The lubrication systems generally provided are as follows:—

1. Thrust Bearing: all types
   - *a*. Oil lubricated: Oil bath or circulating system
   - *b*. Grease lubricated: Grease filling.

2. Guide bearings:
   - *a*. Oil lubricated : Circulating system
   - *b*. Grease lubricated : Grease filling or grease cup

Operating Mechanism for Both Horizontal and Vertical Turbo sets:—

1. Hydraulically operated Governor, servomotor and remote control. Oil is used as the hydraulic medium and very often it is the same oil as is used for the main bearings.

(b) Operating mechanism within the runner head of Kaplan turbines. Oil bath system of lubrication. Generally lubricated by filling the head with oil.

c. Miscellaneous small bearings on the turbines such as pins, shackles slides, gears, etc

Oil lubricated: Hand oiling
Grease lubricated : Grease cup.

Lubrication Requirements of Hydraulic Turbines and how these can be adequately looked after.

The lubricants requirements of hydraulic turbines are not particularly exacting or complex for the number

of parts to be lubricated are few; the conditions of operation not particularly arduous and the scope for design of suitable bearings not too restricted. Accessibility is somewhat restricted as in the case of the control gear in the runner head of Kaplan turbines; contact with water or moisture laden air cannot always be avoided.

A satisfactory water turbine oil must therefore first provide:

1. adequate and dependable lubrication;
2. must be easily able to separate from water;
3. must resist oxidation particularly in view of its use as a hydraulic medium in the governor and allied control gear where it comes in contact with compressed air;
4. resist frothing;
5. possess rust-resisting properties over long periods of service.

A chart of oils which are considered suitable for hydraulic turbine is give below:—

(*a*) For all main bearings oil lubricated:	*Circulating or oil bath system*—Solvent refined paraffinic low viscosity, low pour point oil specially strengthened in respect of its 'Anti-rust', 'Anti-form' 'Anti-oxidant' characteristics and of turbine oil quality., Viscosity range from 80″ to 105″ R. I. at 140°F depening upon makers recommendation.
(*b*) Guide bearings:	Same as for main bearings.
(*c*) Drip feed oilers or ring or collar lubrication	A straight mineral (acid refined) or compounded oil of second quality of viscosity of about 125″ R. I. at 140°F.
(*d*) Ball and roller bearings	Same as for main bearings.
(*e*) Oil filled propeller head of Kaplan turbine.	Paraffinic solvent refined straight mineral oil of a viscosity about 650″ R.I. at 140°F.
(*f*) All grease lubricated parts:	For grease cup or grease filling. A calcium base grease of NLGI No. 3 or a Lithium base grease of NLGI. No. 3.
(*g*) Grease lubricated Ball and Roller Bearings:	A high quality soda base grease of NLGI No. 3 if the bearing does not come in contact with water or a calcium or Lithium base grease of NLGI. No. 3

Even the best lubricant is not invulnerable and therefore in order to obtain satisfactory and economical results from a charge of lubricant the undernoted precautions are worth taking in respect of hydraulic turbine installations.

1. *Storage:*—The oils and greases must be properly stored so as to prevent contamination of any kind and also exposure to heat of any kind, in storage. The oil containers must be properly marked so as to obviate any risk of charging a bearing with any but the correct grade of lubricant.

2. *Defects in plant:*—The performance of the lubricant will obviously be dependent on the mechanical condition of the turbine as a whole and upon that of bearings and the lubrication system in particular. These should be maintained in proper condition. For instance air leaks may cause foaming, water leakage may cause emulsification and sluggish flow of oil.

3. *Incorrect size or alignment of pipes:*—Pipes should be carried in such a way as to prevent aeration. Oil pipes should be adequate in size so as not to restrict flow at any stage; If connected to gas loaded accumulators, pipes must be so arranged as to allow for adequate rate if air release on the L. P. side to prevent air entrainment.

4. *Impurities in the system:*—All impurities and adventitous matter should be removed from the oil by proper purification. A purifier may be included in the oil circulating system and the oil may be purified periodically by batch purification.

5. *Coolers:*—Oil coolers must be adequate in size and well maintained so as to achieve efficient cooling.

6. *Periodic checking of the condition of the oil in circulation:*—

It would be good practice to have the oil checked periodically for the following characteristics.

Specific Gravity;

Flash point;

Viscosity;

Demulsibility;

Total acidity.

A Report on a Design and Development of a Model Francis Turbine

A brief report on the design features and performance of a Francis Turbine made at the P. S. G. College of Technology is presented in this paper.

By G. R. DAMODARAN
and K. A. ABDUL RAHIMAN

SPECIFICATION

The output and total head of the Francis Turbine were selected to be 4.3 H.P. and 48 ft. head of water respectively, taking into consideration the availability of the testing equipment in the hydraulic laboratory. As we have plans to make reaction turbines of different specific speeds, we selected a value of 25 to this unit. The speed of the turbine is a synchronous speed of 1500 r. p. m.

RUNNER DESIGN

Assuming an overall efficiency of 75%, discharge was calculated to be 400 g.p.m, the following empirical relations have been used in the design of the Francis runner.

$$\frac{U_1}{\sqrt{2gH}} = 0.7. \quad \frac{B}{D_1} = 0.15. \quad \frac{D_2}{D_1} = 0.8$$

where H = Head in feet of water
U_1 = Peripheral velocity at inlet in FPS.
B = Height of the guide vane.
D_1 = Diameter of runner at inlet.
D_2 = Diameter of runner at outlet.

The number of vanes has been taken as ten.

The number of guide vanes has been selected as eight, in order to prevent periodic flow-rate variations when the runner vanes pass over the guide vanes. The number of stay vanes or speed vanes is also made equal to that of guide vanes.

Discharge is axial and the entry is radial for the given specification. The velocity of flow is 0.15 $\sqrt{2gH}$ and speed ring vane angle is 12°6′.

The shape of the runner is as shown in Fig. 2. The runner is keyed to the shaft and is held in position by a stream-lined nut, so that the curvature of the inner contour is uniform throughout.

DEVELOPMENT OF VANES

The inlet and outlet angles being fixed, the shape of the vane between these points has to be designed to give steady motion to attain maximum efficiency. Hence it has been done such that the passages are nowhere divergent, and that any change of curvature is as gradual as possible.

The runner space is divided into four stream tubes having equal flow rate, using the method of "equal area construction". Then each of the five flow lines (including the inner and outer contours) is developed to give the true length on paper by error triangle method, where again the curves are drawn by mid-ordinate construction.

Then the plan view is developed by circular projection. The plan views of all the five lines are arranged such that the inlet edge is inclined parallel to the axis by 10°. The outlet ends of all the curves in plan view are joined by a smooth curve which forms the outlet edge of the vane. Then the runner, in elevation, is cut by a number of planes perpendicular to the axis, at intervals of 1/8″. The lines of intersection of the planes and the vane are then projected on to the plan view to get the pattern views. The shape of the pattern views cut from 1/8″ planks when piled one over the other gives the surface of the vane. From the wooden pattern aluminium vane patterns are made.

The thickness of vane was taken as 1/8″ giving consideration to the workability during moulding. With the co-operation of the skilled staff of the P. S. G. Industrial Institute, it was possible to make a runner with smooth vanes pertaining to the designed shape.

DESIGN OF OTHER PARTS

Spiral casing:—A circular section is used and the velocity of water over the entire periphery is kept constant at a value of $0.15\sqrt{2gH}$. Hence the area at any cross section decreases proportional to the angular displacement of that cross-section from the inlet of the spiral casing. The casing is cast of $\frac{1}{2}$″ thick cast iron.

Guide vanes:—The shape of the guide vane is such that it will produce free vortex, guiding the water through the designed angle. The losses due to shock at entry and flow through the duct are minimised by streamlining the vanes. The eight gun metal guide vanes are controlled by a guide ring assembly.

Assembly:—The two inch diameter shaft is of mild steel and is lined with gun metal bush where it comes in contact with water to avoid corrosion. The loading of the turbine is done by rope brake on a brake drum of 12 inches diameter. Two ball bearings capable of taking end thrust are used. The sectional assembly is shown in Fig. 1.

Results and conclusion:— The characteristic curve is given in Fig. 5. The maximum efficiency of the unit

A Report on a Design and Development of a Model Francis Turbine

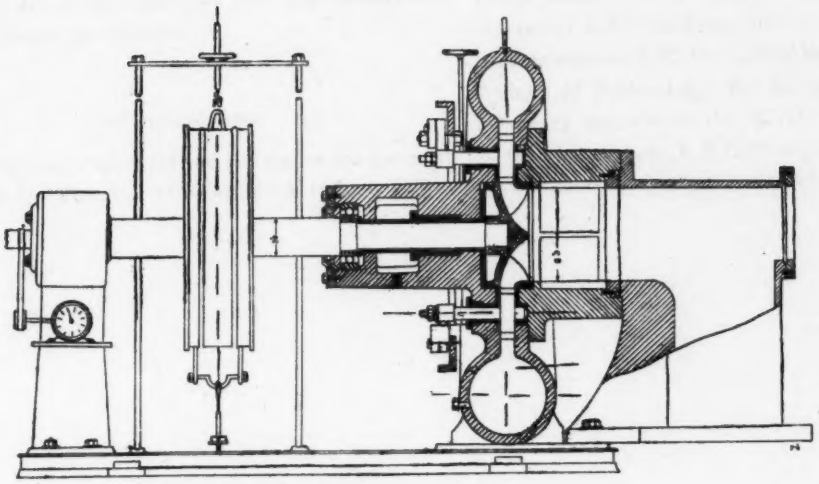
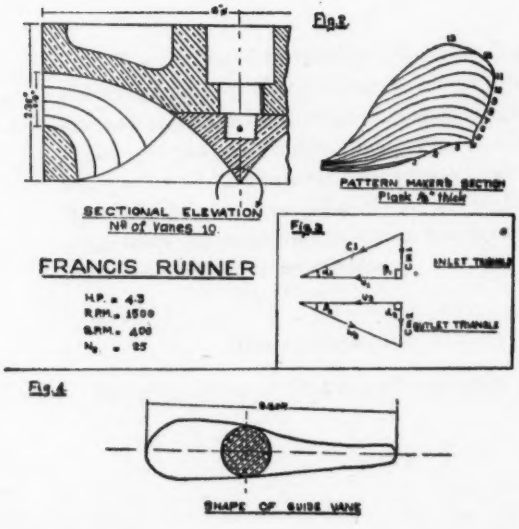


FIG 1. FRANCIS TURBINE. $N_s = 25$.
H = 48' Q = 400 g.p.m. N = 1500 r.p.m. P = 4·3 H.P.



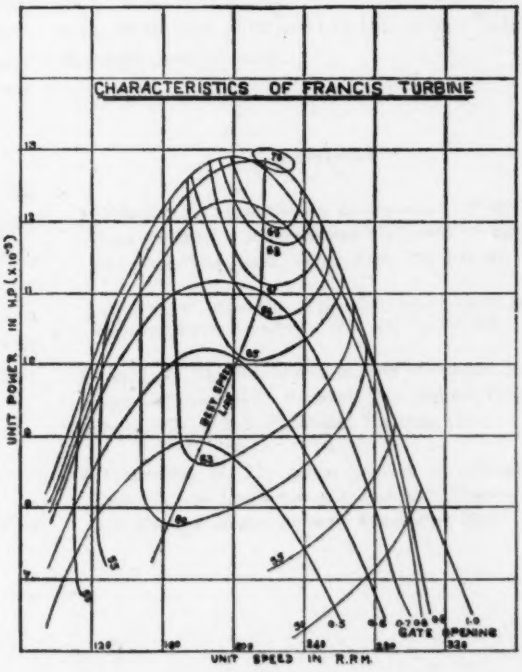


Fig. 5

is 71%. But for this low efficiency, the operation of the turbine is very close to the design specifications. With the experience gained, it is hoped that the efficiency of the other units now under construction will be comparable to any other foreign make. For testing we made use of the Universal Test Rig available in our hydraulic laboratory.

Acknowledgement

This scheme is being carried out under the financial aid from the Council of Scientific & Industrial Research, New Delhi. The design and the production of the unit in such a short time was possible due to the close collaboration of Sri. R. K. K. R. Govindarajan (Asst. Prof. in Mechanical Engineering, P.S.G. College of Technology) and his constant inspection during every phase of work from the drawing table to the assembly. Acknowledgements are also due to Sri. S. Nagaratnam (Asst. Prof., Civil Engineering, P. S. G. College of Technology) for his guidance whenever necessary and also to Sri. K. M. Srinivasan (Senior Research Assistant, P. S. G. College of Technology) in assisting during erection and tests.

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Effect of Blade Number on the Torque/Slip Characteristics of a Small Fluid Coupling

4. Variations in the number of blades in a coupling considerably affect the torque/slip characteristic. The optimum number of blades appear to be fairly critical and gives both the best torque characteristic and the least "shock" losses.

5. It is confirmed by results of the N. E. L. and full size couplings, that the theoretical analysis by Oprecht gives an approximate method of determining the main characteristics of a fluid coupling. The results from the small N. E. L. coupling show greater scatter than Oprecht's results, due probably to the wider scope of the tests. An empirical correction has been suggested to correct for these effects.

Acknowledgements

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Report of the Discussion

By N. S. GOVINDA RAO

The 48 papers received were discussed in five sessions. *During the first Session* 16 *papers received on the subject of Hydraulic Turbines were discussed.* This Session was presided over by Sri. P. Govindakrishnayya. Principal, College of Engg. Anantapur.

It was generally felt that steps should be taken immediately to initiate research on fundamental and basic problems pertaining to hydraulic machines in the country. It was also felt that steps should be taken to standardize acceptance tests turbines by model testing in laboratories outside those of the manufacturer. The purchasers of the hydraulic turbines could assist the research workers in requesting the manufacturers to supply the model of the turbine they were buying. There was very little scope for further improving the efficiencies. But their sizes could be made smaller if more research work on cavitation was undertaken. The size of the turines should be limited to the transporting capacity of our railways and roads. As far as possible the size of the units specified should be such as to be capable of being manufactured in the country itself. A certain amount of standardization was necessary in some of the equipment in power houses to facilitate their manufacture in the country and to facilitate interchangability of parts. So far as design of draft tubes were concerned, it could be best done at the various hydraulic research stations established in several states of the country. This should not be left to the manufacturer, who being very far away from the power site could not completely visualize all the flow conditions under which draft tubes made had to work both before and after the plant was installed. While the manufacturers could design their draft tubes on the basis of data supplied to them their designs should further be scrutinized by model studies in the local hydraulic research stations. The new types of turbines of Mr. Deriaz and Mr. P. Danel were full of promise and should be used under conditions where they could work efficiently. Conditions in India have still not come to a stage where pump storage schemes could be thought of seriously. However, it was felt that preliminary studies could be usefully made regarding power generation from tidal estuaries. The papers on the design of Model turbines attracted attention.

Sixteen Papers were received for the Second Session on Pumps. This Session was presided over by Sri. Lalchandani, Chief and Hydraulic Engineer, Jyoti Pumps, Ltd., Baroda.

Similar feelings were also voiced that a hydraulic research centre was an urgent necessity for the country. The existing hydraulic machines laboratory of the Indian Institute of Science should be encouraged to make a beginning in hydraulic research by starting a separate unit for this purpose. A few of the manufacturers representatives present felt that the research unit should be capable of not only doing development work but it should also be able to furnish designs to manufacturers when requested. Prof. N. S. Govinda Rao described in detail the new theory for designing axial flow pumps. He described the characteristics of the impellers designed to produce vortices which have a function between a free and a forced vortex. Sri Lalchandani dealt with the input aspect of a centrifugal pump. It was felt that the studies made regarding prevention of vortices pumps were very useful. A paper on the high discharge low head pump designed at the Institute evoked interesting discussion.

The third Session was devoted to discussions on hydro dynamics of turbo machines. This Session was presided over by Professor N. S. Govinda Rao, Indian Institute of Science.

The various stages of cavitation in hydraulic machines were discussed. Attempts made at theoretical estimates of the extent of cavitation likely to occur in the proto-type of a machine were largely yet in preliminary stages. Further researe work was necessary. The discussions centred round the physical causes of cavitation, the causes of inception of cavitation and the methods of predicting cavitation effects on prototypes from model studies. It was felt that research work should be concentrated on predicting cavitation on the various parts of a pump individually and collectively and study their characteristics on the efficiency and performance of pumps.

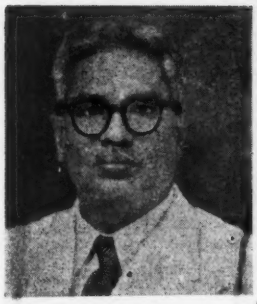
In the Fourth Session six papers were discussed on Testing of Hydraulic Machines and other miscellaneous subjects. This Session was presided over by Prof. N. S. Govinda Rao.

It was felt that serious attempts should be made to standardize all sizes of pumps as have been done in Germany and other countries. Standardization of turbines should also be undertaken. Electronic type of governors were considered more suitable than the hydraulic type for operations of large grid systems. The importance of initiating research in this very important field of automation in the power generation industry was felt.

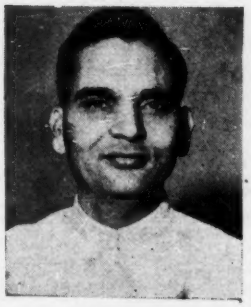
The paper on Design of Foundations for Heavy Machines was welcomed.

The Session closed with a vote of thanks to the several Sectional Chairmen and to the various delegates who took part in the Symposium.

Our Contributors



Dr. S. Bhagavantam is Director, Indian Institute of Science since March 1957. Education: Nizam College, Hyderabad, Indian Association for the Cultivation of Science, Calcutta; Professor of Physics and Principal, Andhra University, Waltair, 1932-48; Indian Scientific Liaison Officer, B.C.S.O., London, 1948-49, Director, Physical Laboratories, Osmania University, Hyderabad, 1949-52; President, Physics Section, Indian Science Congress, 1946; Visited China as the Scientist Member of the Cultural Delegation sent to that country by the Indian Government, 1952; Convener, General Education Team appointed by the Government of India to visit U. S. A. and U. K., 1956; Vice-Chancellor, Osmania University, 1952-57; Visited Russia as a member of the Delegation of Scientists sponsored by the Government of India, 1958. Foundation Fellow and now Vice-President of the Indian Academy of Sciences. Publications: Several scientific papers and two books, viz. Scattering of Light and Raman Effect, and Theory of Groups and Its Application to Physical Problems.



Mr. Manubhai Shah is Minister of Industry, Government of India.

Mr. V. P. Appadurai joined Madras Electricity Department during 1932. He rose to the position of Superintending Engineer of the Technical Branch in 1948 and became Chief Engineer of the Madras Electricity Department in 1954. After the formation of the State Electricity Board in 1957, he is now the Chief Engineer and Technical Member of the Board.

During 1952 he visited U. S. A., Canada, U. K. and the Continent under the Colombo-Plan Fellowship Award.

During 1956 he went to Canada as a Member of the Govt. of India Delegation to finalise the terms of Colombo Plan Aid for the Kundah Hydro-Electric Scheme. He again went to Canada during 1955 for finalising the orders for plant and machinery required for the Kundah Hydro Project and to Germany during 1958 to expedite the despatch of the generating machinery required for the Periyar Scheme.

Professor N. S. Govinda Rao passed his B.E. (Civil) with distinction in 1928. He has a very distinguished career in the technical educational field and in research. Published over 50 papers on research in hydraulics, irrigation, water power development, conservation and soil mechanics. His investigations include aerofoils, current meter, domestic, water supply meter, high discharge low head centrifugal pumps and low cost transmission towers with pre-stressing device. He has 12 years experience in investigating, design and construction and maintenance of hydraulic and other civil engineering works. He is at present Dean of the Engineering Faculty of the Indian Institute of Science. He is member of various academic and professional bodies such as Institute of Engineers, IAHR, American Society of Civil Engineers, British Hydrodynamics Research Association. He has toured U. S. A. and Europe extensively and represented this country in International technical conferences.

He is the author of several publications including some published by the CB & IP. He is consultant in various projects and member of the Board of Studies in the Universities of Madras, Kerala and Annamalai.

Mr. P. Deriaz Dip. Mech. Eng. (Zurich), was educated at the State School in Geneva and the Federal Institute of Technology in Zurich where he obtained the diploma in hydraulic machinery in 1921. He spent a further year at the Institute as assistant lecturer in mechanics before joining the English Electric Co. in 1922 as first assistant designer in the Hydraulic Department. Mr. Deriaz was appointed chief designer of the Water Turbine Department in 1950, and some of his principal work in recent years has included the design of the most powerful water turbines at present in operation in the world (Chute des Passes,

Canada), the 16 Canadian turbines of the St. Lawrence power house and the ten 130,000 HP Kaplan turbines for Priest Rapids on the Columbia River. His most noted work is connected with the mixed-flow variable-pitch turbines and pumps which bear his name and and came into service in 1957 at the pumped storage scheme of the Sir Adam Beck Niagara Power plant of the Hydro-Electric Power Commission of Ontario.

Mr. R. K. K. R. Govindarajan graduated in Mechanical Engineering from the University of Madras in 1951. He is at present working as an Assistant Professor of Mechanical Engineering in the P. S. G. College of Technology, Coimbatore. He has been engaged in several Research and Development schemes conducted in the Hydraulic Laboratory of the above college. He has served as Investigator-in-charge of the Research Scheme "Design and Development of Venturimeters and Orificemeters" sponsored by the Council of Scientific and Industrial Research. He is at present Investigator-in-charge of the CSIR Scheme "Design and Development M'ixed and Axial Flow Pumps of Moderate Discharge Capacity". He has served as a member of the Sub-committee of the Indian Standards Institution for "Horizontal Spindle Centrifugal Pumps for Clear Cold Fresh Water". He is also serving as a Consulting Engineer for the Design and Development of Pumps and Hydraulic Machinery for the P. S. G. Industrial Institute of Coimbatore.

Mr. N. Rajaratnam took his B. E. degree in Civil Engineering in First class with honours from the College of Engineering, Madras in 1957. Took his Master's degree in advanced hydraulics from the same institution in 1958. Worked for about a year in the Irrigation and Research Station, Poondi. During that period, he has solved a number of problems in applied hydraulics which were published in technical journals, both Indian and foreign. Joined the Hydraulics Section of the Indian Institute of Science, in 1959 as a senior Research Fellow for his Doctorate in "High Velocity Flows in Open Channels" under Prof. N. S. Govinda Rao. Was appointed as Senior Scientific Officer in 1960 in the same Institute. He has published over a dozen original contributions in Indian and foreign journals. He is a member of the International Association for Hydraulic Research.

Mr. Pierre Danel President-Director General, Sogreah, France. Mr. Danel was in charge of the courses at the Ecole des Hydrauliciens de Grenoble and Director of the Laboratoire from 1933-35. He has made extensive studies on many aspects of hydrology the main among which are:

Studies on maritime hydraulic problems (Harbour layout and siting, stability of jetties, perfection of jetties, clappotis, tidal and wave energy).

Investigation of the problems of river hydraulics (the cutting of bends, training, density currents, etc.)

Studies into hydrodynamics and general hydraulics (similitude, etc.)

Hydraulic measurement (modules, Pitot tubes, current meters, weirs, etc.).

Studies on the construction of gates for large dams.
General studies on syphons.
Investigation of arid and semi-arid areas.
Investigation of sub-terranean water.

He has made an important contribution to the whole field of hydraulics as applied to agriculture. In particular he has invented an entire range of equipments allowing automatic distribution in a rational and controlled manner of the water in irrigation network.

He is associated with quite a large number of international technical bodies of which he has been past President.

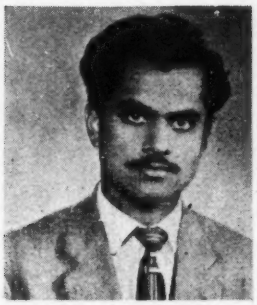
Among his leading publications are:

"Sur la loi de Reynolds et son application aux freins hydrauliques" in 1937 in collaboration with R. Sutterlin.

"La repartition de la charge entre less groupes d'une central hydro-electrique" in 1942 in collaboration with Rebele Pinto.

"Cours d'hydrodynamique graphique" (Studies of conformal transformations based on elementary geometrical considerations for hydrodynamic applications) (one volume published.)

Mr. G. Suryanarayana Iyer had a successful academic career. After completing M. E. Degree in Power Engineering (Civil and Hydraulics) at the Indian Institute of Science, joined the Hindusthan Construction Company Ltd. and presently working in their design office at Bombay.



Mr. A. S. Magaraja studied in National Institute of Engineering, Mysore and secured B.E. (Civil) in 1958 with a First class. Undergone post-graduate course in power engineering (Civil and Hydraulics) at Indian Institute of Science and secured M. E. Degree with distinction. (1958-1960)

Undergone training at C. W. &. P. C. Delhi and presently employed as Assistant Engineer in Assam State Electricity Board.

Mr. L. H. Kessler is Chief Hydraulic Engineer, Fairbanks, Morse & Company and formerly Chief Engineer of the new Kansas City, Kansas Pump Works. He joined the firm in 1952 as Chief Hydraulic Engineer at the Beloit, Wisconsin Works. He had been professor of Hydraulics at the University of Wisconsin for 24 years and holds three engineering degrees from the institution. Durng the war he was on leave to serve as Chief of Water, Sewer and Services, Office Chief of Engineers, War Department, 1941-45. In 1946 he became Professor of Sanitary Engineering, Northwestern University. During his teaching career he was Consulting Engineer on a number of important projects in the middle west. He is a member of National Engineering and Scientific Societies.

Mr. N. S. Lakshmana Rao is a Lecturer in Civil and Hydraulic Engineering at the Indian Institute of Science, Bangalore. He is a graduate in Civil Engineering of the Mysore University, an associate of the Indian Institute of Science, an associate member of the Institution of Engineers (India) and a member of the International Association of Hydraulic Research.

After graduation in 1951, he joined the Indian Institute of Science as a Research Scholar and was awarded the Associateship of the Institute in 1954. He is the first associate of the Institute in Civil Engineering.

From 1954 to 1956, he worked as an Assistant Professor in Civil Engineering at the A. C. College of Engineering and Technology, Karaikudi (Madras State) and built up the hydraulic laboratories of that college.

He was deputed by the Institute under a UNESCO Fellowship to U. K. and France, in October 1958, to familiarise himself with the use of some latest instruments in Hydraulic Research.

Mr. M. W. Lalchandani, B.E. (Elect) B.E. (Mech.) has 22 years experience in Design and Manufacture. For the last 15 years Chief Engineer in Jyoti Ltd., Baroda, he has been responsible for Design and Development of Pumps, Manufacture and Production Techniques etc.

Mr. B. P. Sangal took his M. Sc., Degree in Mathematics from the University of Allahabad in 1954 and thereafter completed parts A and B of Associate Membership Examination of the Institution of Engineers (India). After serving for three years as lecturer he joined the Civil and Hydraulic Engineering Section of the Indian Institute of Science, as a research scholar in 1957 and was subsequently awarded a Research Fellowship of the CSIR. He has been engaged on the problem of Cavitation and has received the M. Sc. degree of the Institute for his thesis "Inception of Cavitation". Recently he has been selected by the University Grants Commission for the award of a Senior Fellowship in Civil Engineering.

Dr. Allan J. Acosta is Associate Professor of Mechanical Engineering at the California Institute of Technology (58) where he graduated B.S. in 1945. M.S. in 1949 and Ph. D. in 1952. He was Assistant Prof. from 54-58. In research he pursues the field of cavitation. Teaches thermodynamics and Hydrodynamics with special interest in turbomachinery; consultant of industries in this field.

Prof. Aladar Hollander Professor Emeritus of Mechanical Engineering at the California Institute of Technology (51) M.E. Royal Tech. University Budapest (04). After a year's study at the Technical University, Zurich (05) worked in Europe and U. S. (05-13) in various fields of engineering.

Settled in U. S. (13) in design of submarine machinery (13-19) then in pumps, which became his speciality; Chief Engineer of LeCourtenay (19-23), of Byron Jackson (23-39), consultant since (39). Joined California Institute of Technology, Lecturer (44) Associate Prof. (45-51) Worked with Dr. Knapp in the Hydraulic Laboratory. Consultant of State of California (52) on the $ 75 billion Feather River Project and on other pumping plans.

Mr. G. Forrest Drake Vice President in charge of Engineering and Research for Woodward Governor Company, graduated from Arsenal Technical High School, Indianapolis, Indiana in 1921. He moved to Rockford, Illinois where he was employed by Barber-Colman Company as a machinist until 1923 From 1923 to 1926 he progressed from draftsman to machine designer in the Experimental Department. In 1926 he entered the University of Illinois graduating in 1930 with a B. S. in Engineering Physics.

Returning to Barber-Colman he was a development engineer until 1934 and chief engineer of the electrical apparatus division from 1944 to 1938. In this year he went with Woodward Governor Company as Chief Experimental Engineer 1938 to 1942, Chief Engineer 1942 to 1954 and Vice President in 1954.

He is the holder of some forty patents on air distribution outlets, air conditioning controls, radio controlled garage door operators, speed governors for prime movers, automatic aircraft engine synchronizers, and associated devices. His published papers include the magazine article "Basic Factors in Applying Speed Governors" published in Machine Design, "The Noise Characteristics of Air Supply Outlets (Ashve) "Governors on Internal Combustion Engines' (Dema) "Governing of Internal Combustion Engines" ASME, OGP Div), "The Development of Gas Turbine Controls" (ASME), and "Speed Control of Prime Movers" (AIEE).

Mr. D. Firth is Head of the Fluid Mechanics Division, National Engineering Laboratory, Glasgow.

Mr. I. S. Pearsall is Senior Scientific Officer, National Engineering Laboratory, Glasgow.

Mr. C. N. Nagaraj studied in Central College, Bangalore, from where he took his B.Sc. degree in 1942. Graduated (in I Class) in Civil Engineering from the University College of Engineering Bangalore. Recipient of a Gold Medal at the University Convocation in 1946. Worked in Mysore P.W.D. for 3 years and was associated with the design and construction of univer- convocation Hall. As Assistant Engineer Construction in the Indian Telephone Industries. he was in charge of design, lay out and excution of industrial buildings, hangars, sanitary and water supply installations and over civil engineering works. Joined the Indian Institute of Science, Bangalore in 1952 for teaching post-graduate students in Power Engineering.

He was deputed to England in 1955 for one year for post-graduate work at the Imperial College, London and for training in vibrations in soils. As Assistant Professor in Soil Mechanics, he is running a Master's Degree course in this subject and is actively connected with guiding, conducting research in the field of soil dynamics i.e. the study of the behaviour of soils under dynamic loads. He has been in charge of the research schemes sponsored by the C.S.I.R. and C. B. I. P. in this field and has published papers connected with foundations of machines.

Dr. Ing. h. c. K. Rutschi is owner and director of the Brugg Pump Factory at Brugg (Switzerland) Academic degree h.c. of the Braunschweig Institute of Technology Technische Hochschule (Prof. Dr. ing. C. Pflederer). He published several hydraulic findings with pumps, such as on cavitation, model testing, standardization and good design. Dr. Rutschi took and active part in discussing hydraulic problems at international meetings for hydraulic research. He is also technical advisor of several pump manufacturers in Europe and overseas, his pump constructions being manufactured under license in 18 countries the world over.

K. Seetharamiah took his B. E. Degree examination of Mysore University in 1935 in Civil Engineering. Certificate of proficiency in Electrical Technology Indian Institute of Science, Bangalore, 1937.

Worked in various capacities in the following hydro-electric projects during 1938. (*a*) Jog power scheme, Govt, of Mysore (*b*) Chambal Hydro-Electric Project (*c*) Radhanagar Hydro Electric Project (*d*) Koyna Hydro-Electric Project. Since July 1951 joined the Indian Institute of Science as Assistant Professor in Civil and Hydraulic Engineering.

Went to U. S. A. under the T. C. M. programme for one year and worked at California Institute of Technology, Pasedena, Los Angeles and at Madison, Wisconsin and U. S. B. R. Denver, Colorado.

Has published more than a dozen papers in Indian journals.

Mr. G. A. J. Young is with the British Hydro-Mechanics Research Association.

Mr. P. Govinda Krishnayya is Principal, College of Engineering, Anantapur (Andhra Pradesh)

Toured extensively in the year 1949 the United States of America and the Continent.

In 1951 the P. S. G. College of Technology was started and Sri. Damodaran became the Principal.

Was Member of Parliament (Lok Sabha) for a period of five years from 1952-57. Was elected to the Madras Legislature Council from the Teachers' Constituency and is serving as a Member from April, 1958.

Went as a member of Top Management Organisation and Training Team sent abroad by the National Productivity Council, Government of India, and toured U. K., West Germany and the United States during September-October, 1959.

Is serving in the following Bodies and Committees Chairman, Board of Studies in Engineering, University of Madras. Member, Syndicate, Senate and Academic Council University of Madras, Member, Electrical and Mechanical Engineering Research Committee, Council of Scientificand Industrial Research, Government of India, Director, Neyveli Lignite Corporation Limited, Neyveli, Member of Institution of Electrical Engineers, London and several other deaing professional institutions.

Mr. G. R. Damodaran was educated in Sarvajana High School, Peelamedu, Coimbatore, Government Arts College, Coimbatore, and King's College, Newcastle-Tyne, Durham University, United Kingdom. Took up B. Sc. Degree in Electric Engineering, and B. Sc. Degree in Mechanical Engineering from the Durham University.

Returned to Indian after study abroad and assumed charge of Director, P. S. G. Industrial Institute, Peelamedu, Coimbatore from 1943.

Mr. K. A. Abdul Rahman studied Applied Electronics and got a Diploma (R. H. A. E. (Grad) of the Institute of Applied Electronics, Calcutta. Served for sometime as Junior Engineer in the Parambikulam—Aliyar Project of the Madras State, P. W. D. For the past one and a half years, he has been engaged as 'Senior Research Assistant' in P. S. G. College of Technology under the scheme "Design and Development of Water Reaction Turbines of Small Capacity" sponsored by the Council of Scientific and Industrial Research Scheme, New Delhi; and doing research under the guidance of Prof. G. R. Damodaran.

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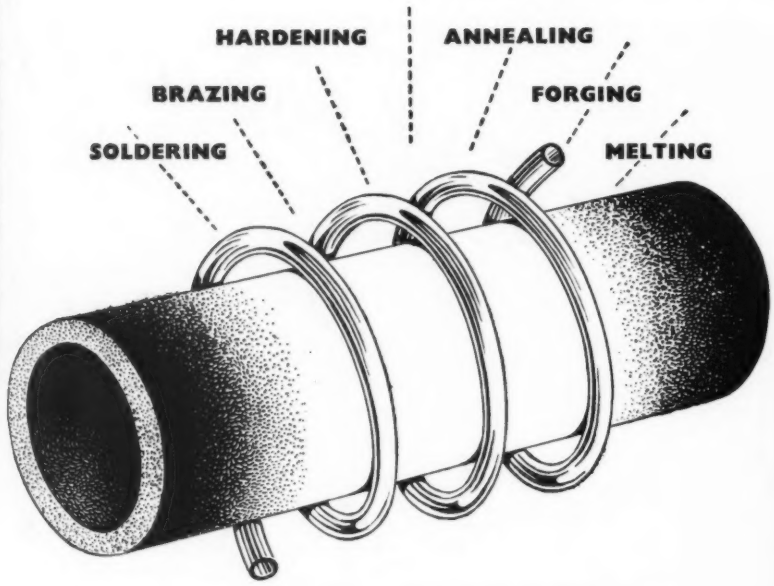

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